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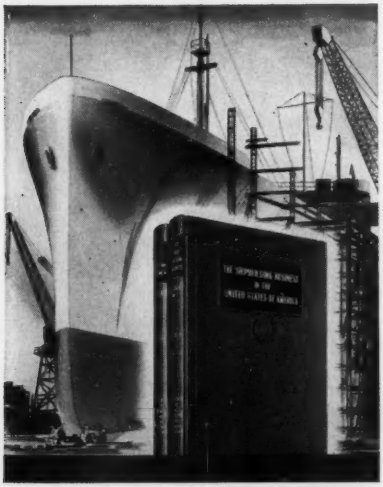

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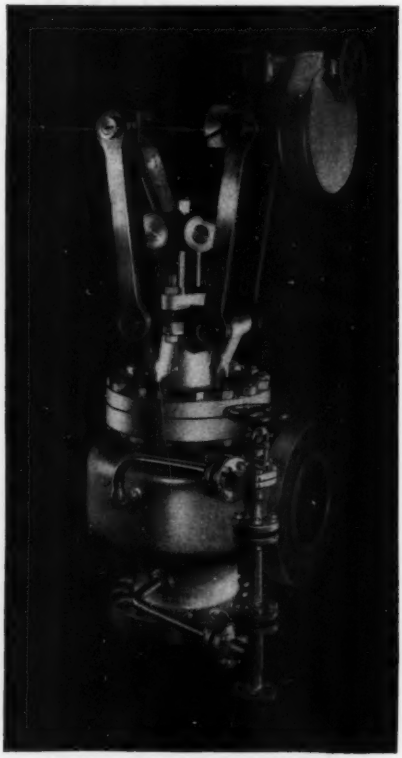

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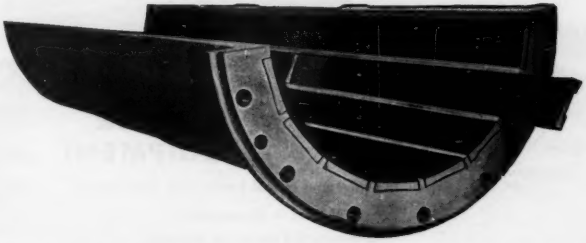

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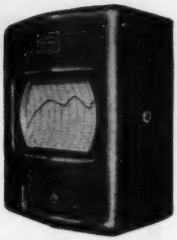

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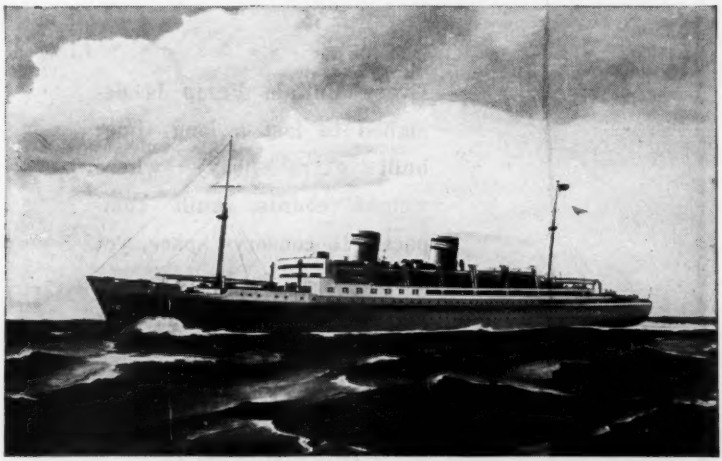

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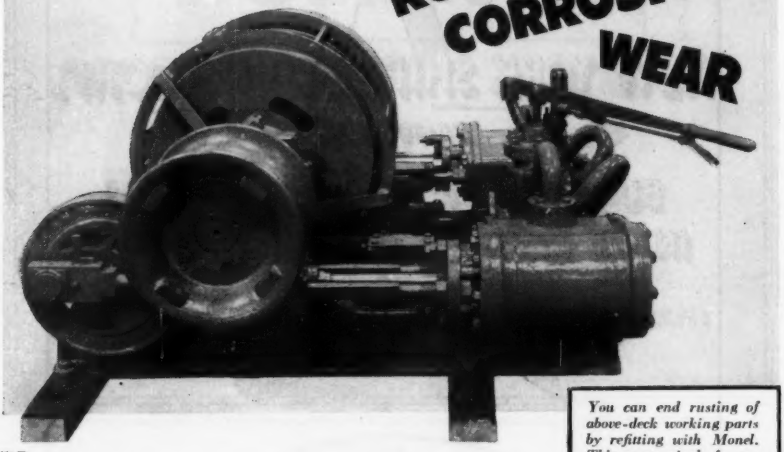

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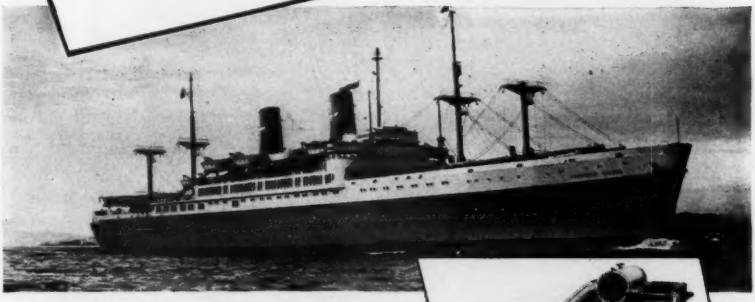

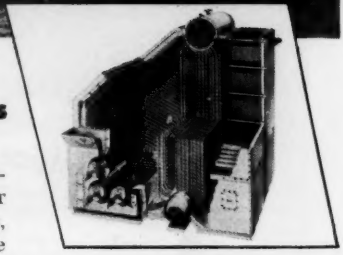

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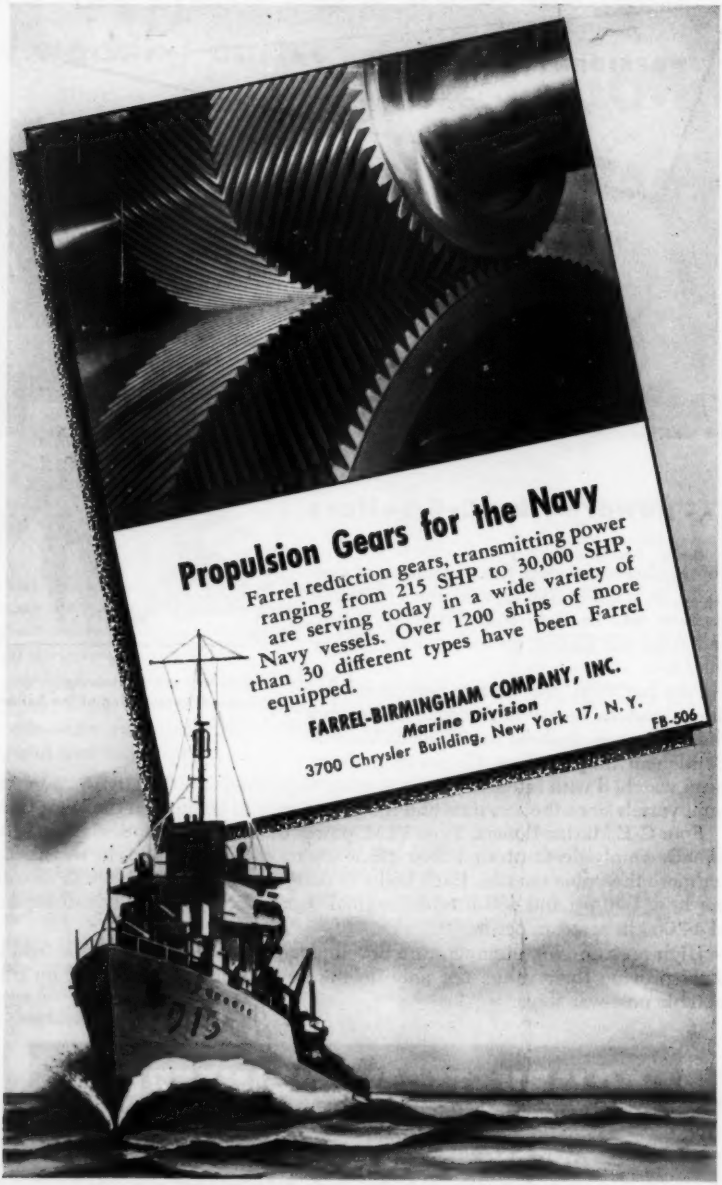
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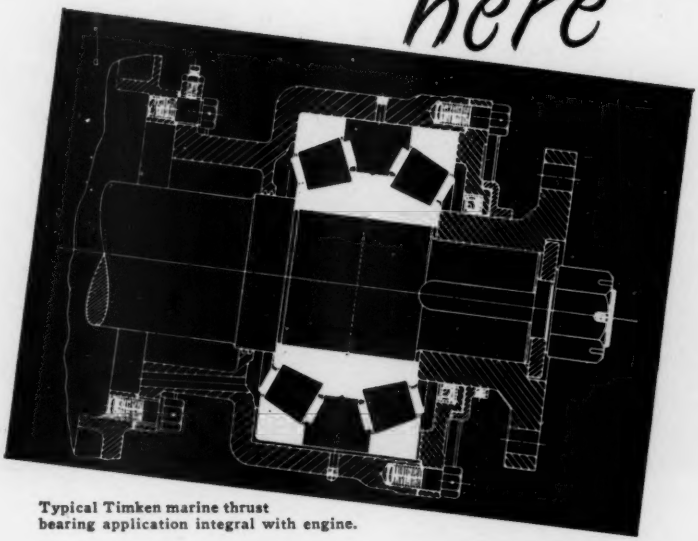

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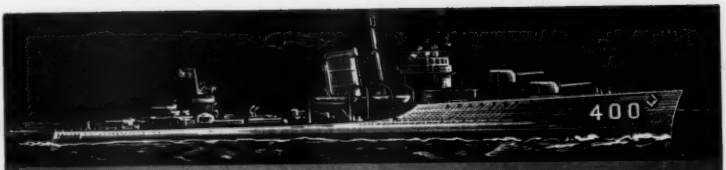

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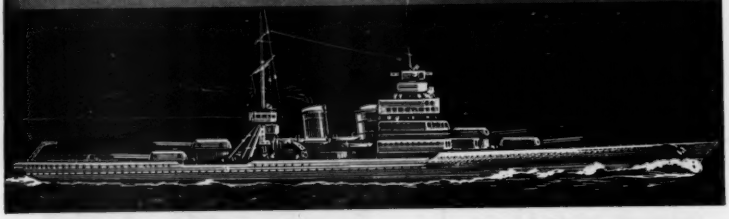

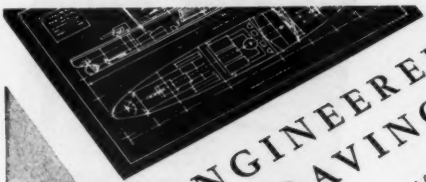

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NATHAN P. TOWNE—The Society's First President in 1889-90. This picture was taken at a later date probably when Chief Engineer Towne was Chief Engineer of the Cramp Shipbuilding Co.

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JOURNAL

OF

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EDITORIAL NOTES

It has been noted that no flood of manuscripts reached the Society as a result of the Editorial Notes which appeared in the August, 1948, issue. Perhaps it is too early to expect results.

The Society has two very real needs, both essential to its health and success. The first of these is articles of the proper quality; the second is a sustained and expanding membership.

The thinking behind the request for manuscripts has been carried in the two previous issues. No new arguments are available to the editor. Only if the JOURNAL offers its readers truly valuable reading material can it successfully carry out its mission. If the members who are qualified as authors are interested in strengthening the Society, they will surely overwork the editorial staff with original contributions.

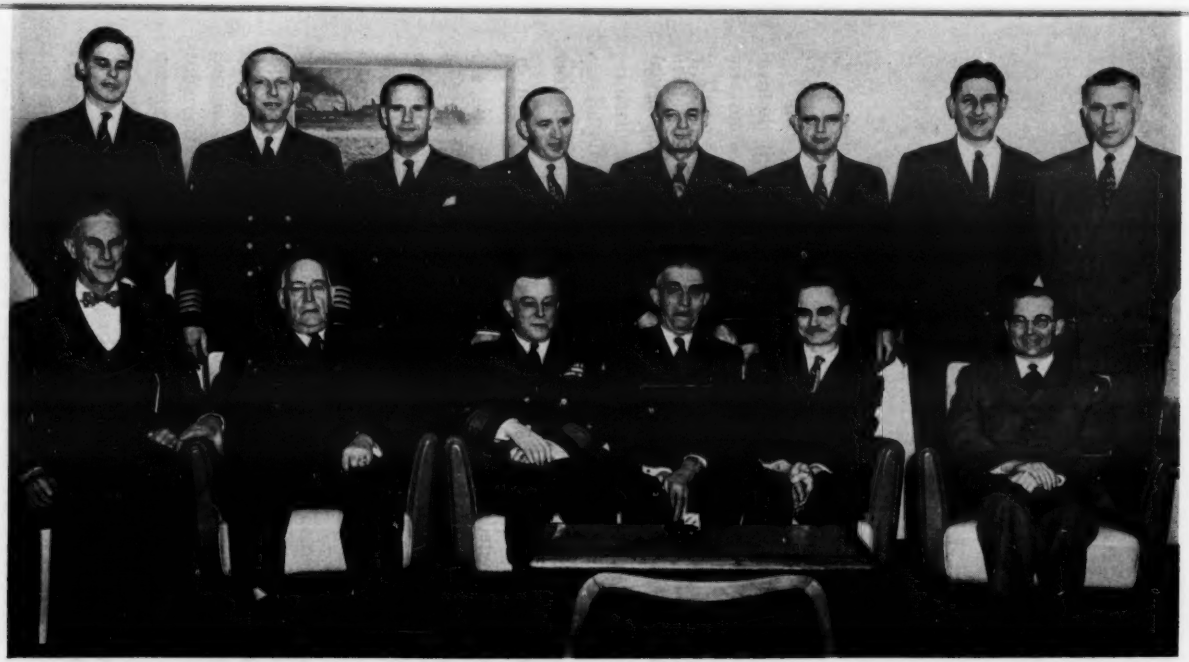
The number of members is very definitely related to the state of the Society's finances. It is probably known to the members that the present reserve is much larger than is required in normal times. It is not as generally recognized perhaps that "normal" does not apply to the present. The current reserve will disappear in 8 to 10 years if the 1948 rate of expenditure and income is continued.

Members are now receiving four JOURNALS a year for the original (1888) membership rate of $5.00. The out-of-pocket cost of these JOURNALS is more than $8.00. This condition has been accepted advisedly by the Council in view of the reserve status. There is plenty of time available to take the necessary corrective action before the Society approaches insolvency. There need be no concern on this score.

Estimates of future costs and predictions of future trends are not possible with scientific or even engineering accuracy. However, based on the best information now available, it appears that a break-even point between publishing costs and circulation would be reached with a paid circulation (members plus subscribers) of between 9,000 and 10,000. This compares with a present paid circulation of about 4,400.

That indicates one possible approach to a solution which will permit continuation of the present $5.00 membership rate and still meet present publication costs. Doubling the circulation will put the Society in a very close to stable financial position. The present members probably have it within their power to make the decision.

As a result of the war, there are at least 300,000 persons eligible as Naval members and probably as many or more as civilian members. One per cent of these added to the rolls would more than suffice to make the practical test of the break-even point. The future of the Society makes that test worth while. During 1949 there will be four application blanks reach each member in the back of his JOURNALS. If these are torn out and put to use by the members, the situation will be cleared up in one year. We would like to leave this matter in the hands of the membership to augment staff efforts to increase circulation, and let the results speak for themselves.



JOINT 1948-1949 COUNCIL MEETING.

Sitting: Capt. Mansfield, 1948-49 Council; Mr. Fessenden, Clerk; R. Adm. Solberg, 1949 President; R. Adm. Johnson, 1948 President; Capt. Hamilton, Sec.-Treas.; Mr. Cooper, 1948 Council.

Standing: Comdr. Meigs, Asst. Sec.-Treas.; Capt. Mumma, 1948 Council; Mr. Gibson, 1948-49 Council; Capt. Dantzler, 1949-50 Council; Capt. Rex Hicks, 1948 Council; Capt. Geo. Hicks, 1949-50 Council; Capt. Ambrose, 1949-50 Council; Lt. Com. Tiedeman, 1949-50 Council.

Absent: R. Adm. Lonnquest, 1948-49 Council; Capt. Gold, 1948-49 Council; Mr. Wilson, 1949-50 Council, and Comdr. Spowers, 1948 Council.

PART I.

(Original Articles)

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part I are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

ENGINEERING REPORTS AS COMPLETED STAFF WORK.

By S. E. McCrary.

Introduction.

"Since a report is fundamentally the communication of information or counsel which is desired and which will be used by someone for a particular end, its success depends primarily on its being planned carefully to meet all the conditions under which it is to serve. The introduction should present a clear statement of the subject, of the purpose, and the plan of organization of the material treated. It often also includes a brief statement of the conclusion, or recommendation, and sometimes a summary of the results or findings. The conclusion should agree and balance logically with this introduction. The body of the report should be so written as to make evident the structural design announced in the introduction. Ideally a report should read coherently and smoothly, and should bear evidence of the writer's mastery of his subject matter in its larger aspects, as well as in its minutest details."*

Translated into Naval parlance, this means that a report should represent "Completed Staff Work". On this premise, the preparation and use of technical reports by professional engineers in the

The Author.

Mr. S. E. McCrary is a Research Coordinator in the Program Planning Branch, Bureau of Ships. He joined the Bureau in 1940 after having served for five years with General Electric Company as Installation Supervisor specializing in Marine and heavy industrial equipment. Upon graduation from Virginia Military Institute in 1930 he became a Student Engineer with General Electric Company and later served for four years as a First Lieutenant of Cavalry. He is now Secretary of the Senior Engineers' Association of the Bureau of Ships and a member of the Recreation Commission of the City of Falls Church, Virginia.

*By permission from "Writing the Technical Report", by J. Raleigh Nelson, Copyrighted, 1940, 1947, McGraw-Hill Book Company, Incorporated.

Naval service will be treated in three parts. First, fundamental considerations; second, specific suggestions regarding report preparation; and finally suggestions regarding the design and use of three kinds of reports for economical and efficient administrative purposes. Routine reports of data and laboratory tests are regarded as special types which will not be discussed.

Fundamental Considerations.

Initiation of any project or task involving expenditure of government funds should not be undertaken except upon advice of competent professional authority on the subject involved. If this be accepted, and indeed it must, for it is one of the underlying reasons for employment of professional people by the government, then it is incumbent upon the engineer who would initiate projects involving public funds to set forth clearly, and in advance, his professional recommendations. He should, furthermore, record and report his professional efforts so that those who succeed him may profit from his successes and avoid his failures.

To be useful, his professional advice and accomplishments must be communicated to others, and this means they must be put into writing, unless they pertain to some extremely urgent, or simple, or incidental matter. An engineering project is rarely simple or incidental, and however urgent, it must be coordinated with other related projects in terms of time, money, personnel, facilities, and many other factors. It is, therefore, an important duty of an engineer to become proficient in designing reports pertaining to his work so that his colleagues may be enlisted for concerted attack.

Engineers are, by popular reputation at least, methodical, analytical, logical, accurate, and skillful in the organization of physical forces or structures. With such qualities, engineers should be able to design reports with relative ease to carry out specified functions. Indeed, an engineer who does not or cannot produce useful reports is in great danger of being misunderstood, and is likely to be regarded as lacking in those qualities which make for professional ability. It is significant, at least, that those engineers highest in the esteem of their collegues are very often those who have developed high proficiency in the art of effective oral and written expression.

Now, assuming that engineers do possess a great store of technical knowledge; that they are not averse to giving others the benefit of it either freely or for hire; that they are equipped with a fair vocabulary and knowledge of English grammar; what simple guides can be set down to help them communicate their ideas and knowledge to others with effectiveness and least burden to themselves?

Specific Suggestions.

There are four important phases in writing a report: study, planning, writing, and self-criticism. First is study of the subject. This requires rigid self-discipline in the process of logical thinking. Engineers are popularly supposed to do this automatically and without conscious effort, but familiarity with this important process is very likely to be the cause of one's undoing. It is quite easy to confuse ideas with facts, and to allow assumptions to become prejudices. It is easy also to allow personal pride or jealousy to obscure objective thinking. It is important, therefore, to guard against these faults of thinking, and to superintend the process so as to obtain the highest possible degree of objectivity.

This study by a systematic thought process is known as the "Scientific Method", or, to the philosopher and to the military man, as an "Estimate of the Situation". It usually involves five distinct steps of fixed sequence and interrelation, each of which must be given incisive treatment. It is the same step-by-step method engineers habitually use in solving problems in mathematics.

First—*Recognition of the mission, problem, or objective.* One should be able to write out a brief but specific and incisive statement of the proposition he intends to cover and why he is doing it. This is a most important step for it is the keynote or basic premise upon which all else depends. It has been said that a problem well stated is half solved. If this be true, then it should be worth a great deal of one's time to state the objective carefully, briefly, and specifically.

Second—*Statement of the situation.* In this step one sets down all the circumstances, all the positively known facts, and all the ideas or suppositions which occur to him as having some possible

influence on the choice of action to be taken with respect to the problem. This step must be treated incisively too, in order to avoid confusion with the proposition as stated, and to exclude material which properly belongs in later steps.

Third—*Analysis of the situation.* In this step a full discussion is made of the interrelation of all the factors set down in previous step, and an estimate is made of the probable outcome of each of the lines of action taken under consideration. Incisive treatment is needed for this step also, to exclude factors belonging to previous steps, for to enter new factors at this step would bring confusion and be evidence of not having covered the previous steps thoroughly. Care should be taken too, to exclude material which is properly a part of the fourth and fifth steps.

Fourth—*Analysis of the lines of action* in relation to their effect upon the proposition as stated. In this step a full discussion is made of the relative merits of the several possible lines of action and the reasons are given for the adoption of one in particular, and the rejection of all others.

Fifth—*Statement of the decision.* This is a brief but completely specific statement of the conclusions reached or the action recommended. It should be a statement of the decision as to *what* should be done and *when, where, how,* and *why,* in that order. This is a very important statement and great care should be taken to insure that it is brief, clear, concise, and complete.

This first phase, the "Estimate of the Situation", in five steps representing a rigorous mathematical logic, may range from mere mental notes to a formal written paper, depending upon the needs of the individual and the complexity or scope of the situation. In any event, the study must be made in all its essentials, however rapidly or briefly, for without it the decision would be intuitive at best and rather precarious basis upon which to state an engineering fact or to plan an engineering proposal. It cannot be said that study of the subject is complete and fully in hand unless it can be reduced to a decisive statement which reflects a logical analysis of all the factors involved.

The second phase is planning the report to meet definite functional requirements. From the viewpoint of the potential reader the report should represent "Completed Staff Work". The writer,

therefore, should strive to make the report a document useful of itself. The plan or structure of the report should be clearly conceived, and before writing a single line, the format, arrangement of material, tone, and literary style should be decided upon. What scheme will logically, practically, and strategically project the subject matter to the mind and hand of the reader in most convincing and usable form? The answer to this question requires an estimate of the situation as used in the first phase, but with the object of reaching a decision as to who is going to receive the report and how it can be made to best serve his purposes. This phase may be considered complete only when the introduction can be drafted in its final form as an accurate synopsis of the report.

The third phase, the actual writing of the report, should not be attempted without thorough study and planning of phases one and two. Adopting this as a rigid rule will help to avoid false starts and reduce the chances of becoming lost in a maze of complex details. Surely, expression comes more easily when one knows exactly what he wants to say, why he needs to say it, and how he ought to say it. Fortunately, much of the material needed for the report in its final form already will have been assembled, either mentally, or in written detail as a result of phase one. Phase three is simply the task of putting it down according to the plan developed in phase two and announced in the introduction.

The fourth phase is self-criticism. Primarily, this is an analysis of the report from the viewpoint of the reader or user, and the adjustment of it to meet his requirements. Care should be taken in this phase to make sure that the body agrees with the introduction. The report should be judged as a document in relation to its usefulness; accuracy and validity of the ideas concerning the subject matter are now of secondary importance. If questions as to the latter do arise, it is clear evidence of having attempted to write without sufficient preparation. This final phase may be considered as complete only when the writer is ready to place his signature upon the report as evidence of professional responsibility.

These four phases, study, planning, writing, and self-criticism, if applied in their logical sequence, provide a systematic approach to report preparation by which one's progress may be guided and checked effectively.

Suggestions on Design and Use of Reports.

An engineering report setting forth some professional advice, opinion, proposition, or record becomes most useful when it is in such form as to be consistent with the doctrine of "Completed Staff Work". Under this doctrine, the report is designed so that the reader or recipient can readily understand its significance, and can accept or reject its proposals, take action upon them himself, or refer the report to someone else for action. A properly designed report will lend itself to being used, in whole or in part, as an authoritative document. In most cases a simple endorsement should be sufficient to authorize the execution of the proposal or otherwise dispose of it. To put the proposition simply: a good report is a useful document.

On this premise, three kinds of reports designed for technical as well as administrative usefulness will be outlined, and suggestions will be given as to how they may be used to save the time of both the engineer and the administrative echelon.

Case I—Engineer's Estimate of Situation.

A report recommending establishment of a program, project, mission, or problem to be solved.

Outline—

Introduction (Synopsis or Abstract)
I Mission or Objective (Assigned or assumed)
II Situation (Known and assumed facts)
III Analysis of situation (Development of possible Lines of action).
IV Analysis of lines of action (Risks and Costs vs. probable success).
V Decision (Recommended action).
Signature and Title of Engineer
Appendix—references (Historical and Authoritative).

Such a report, if properly written, will inspire confidence of the reader in the writer's mastery of the situation, and lead the reader to accept the validity of the decision. By reading only the introduction he will know whether he needs to read further. If he needs further details the introduction will tell him in what part of the report they may be found. If the decision recommends com-

mitments of funds, operating forces, or industrial facilities, and the reader has authority to make them, he can do so or deny them with full appreciation of the consequences. By a simple endorsement such a report can be either rejected or converted into an order upon someone to carry out the recommendations.

CASE II—ENGINEER'S PLAN OF ATTACK.

A report recommending specific action required to solve a problem, conduct an experiment, undertake construction, or accomplish some definite mission.

Outline—

Introduction (Synopsis or Abstract).
I Situation (Known and assumed facts).
II Objective (Based upon a decision reached through an Estimate of the Situation as from a Case I report).
III Plan of Attack (Specific details to be executed).
IV Administration (Ways and means of financing and coordinating).
V Reports required (Frequency, form, and distribution).
Signature and Title of Engineer.
Appendix—references (Historical and Authoritative).

This type of report should be predicated upon and specifically refer to a Case I report if such exists. It will present to other engineers and to administrative or management officials a definite plan which they can readily understand, and can accept or reject with full appreciation of the consequences. When endorsed by competent authority the report is converted into an order or specific directive upon someone to do a job.

CASE III—ENGINEER'S REPORT OF MISSION COMPLETED.

A report for record and reference giving an account of an engineering program, project, or experiment.

Outline—

Introduction (Synopsis or Abstract).
I Situation (Known facts as from Case I and Case II Reports).
II Objective (The task accepted or assigned).
III Plan of attack (Details of execution).

IV Results (Data and observations).
V Conclusion (Recommendations).
Signature and Title of Engineer.
Appendix—statistical analysis of data-references (Historical and Authoritative).

This type of report should be predicated upon and refer to Case I and Case II reports if they exist. It provides technical information on procedures and gives quantitative data which can be used in the solution of future engineering problems. It is a record of successes to be followed, as well as one of failures to be avoided. It can be converted into official authoritative information by simple endorsement.

These report outlines have been designed as guides for producing documentary units which lend themselves to being used in "Completed Staff Work". Each begins with an introduction which should, in less than 15 seconds reading time, let the recipient decide whether it is his business to read further. Systematic presentation of subject matter as suggested by these outlines makes it possible for him to know where different kinds of pertinent details may be found. Whatever action the reader may be required to take can be done with confidence that springs from being well informed. It is certainly not suggested that report writing be superimposed upon one's present work burden, but rather that good report writing be applied as an improvement upon one's manner of doing his job. It is suggested that the doctrine of completed staff work can be applied to engineering communications with advantages to both the engineers who write them and to the employers or clients who read them.

Conclusion.

One or another of these suggested outlines can be adapted to meet nearly every situation which requires the communication of technical information. They need not be followed rigidly in every detail, but whatever outline is used should impress the reader with the fact that the report has been carefully prepared by a competent engineer and designed to be administratively useful. This should be true whether the report is written or oral; a single sentence or an entire volume.

By so applying the principles of "Completed Staff Work", the engineer may relieve himself of much arduous letter writing and paper work. Through such reports, and by his signature upon them as evidence of professional responsibility, he will advance his prestige (or lose it) and have tangible proof of technical contributions in which his employer may take just pride.

However much technical information a man may have, it will not qualify him as a professional engineer unless he applies the "Scientific Method" to "Completed Staff Work" as his everyday professional product.

THE U. S. NAVY PLASTIC BOAT PROGRAM.

By Commander A. C. Bushey, Jr., U.S.N.R.

The difficulty of procuring sufficient quantities of plywood and well seasoned, air dried lumber for an emergency boat building program led the Bureau of Ships to investigate the possibility of utilization of other materials. Metals have been used successfully for small boat construction, e.g. in lifeboats used by the Merchant Marine. Nevertheless, due to weight restrictions, the best metallic boats have, in spite of stiffening incorporated in the manufacturing process, had such thin skins as to make them unacceptable for Navy use. As metals are critical in times of emergency, consideration was given to the newest material, plastic. All of the plastic materials considered are produced from native and readily available raw materials, the output of the finished product being limited only by the plant capacities of a new industry. For boat building, it appeared that a finished boat of plastic would weigh less than either a wood or a metal boat of the same dimensions, that there would be less framing and that the skin panel strength would be greater. A plastic boat would also be a monolithic structure easily repaired if damaged, not subject to those weaknesses of timber construction—failure of fasteners on account of fatigue around screws, nails and bolts; dimensional instability and distortion, dry rot and borer activity.

The Author.

Commander Bushey is stationed in the Bureau of Ships, Contract Design (Hull). He was commissioned 28 February 1930 as a Lt. (J.G.)—Construction Corps, U. S. Naval Reserve. On 7 October, 1940 he was called to active duty and assigned to the Design Section of the Norfolk Navy Yard. To aid the training of personnel in that rapidly expanding organization, he organized the University of Virginia Norfolk Extension for Naval Architecture, teaching in the evenings for two years. He was next assigned to the Pacific. He holds the Bronze Star Medal. Recently the King of England appointed him an O.B.E. in recognition of his services as officer in charge of shipbuilding and ship repairs in New Zealand and commanding officer of the U. S. forces in Auckland.

Investigation also revealed the tremendous savings in equipment, time and money, if in another emergency boat building program, plastic fabrication took precedence over the traditional material and construction method.

Accordingly, in June of 1946, the Bureau of Ships placed two development contracts, each for the construction of three 28′ laminated plastic boats. (Length O.A. 28′ 11″, Beam (Molded) 10′ - 3″, Depth 6′ - 10″). On 30 December 1947, the first boat was lifted from the mold, the integral hull and decking completed in all respects, including equivalent of painting. It was made in 24 hours 10 men participating. (The third hull was molded in less time—about 16 hours.)

Fabrication of this boat required the following equipment:

(1) A polished female hull mold of one-half inch thick aluminum plate, supported by a heavy welded steel basket in which a heating element composed of ¾″ steam pipes on about 6″ centers was set against the aluminum mold, the basketwork resting in a concrete pit (foundation) so that the deck edge was coincident with the shop floor;

(2) A polished female deck mold of ½″ aluminum, backed by a heavy steel structure, which likewise contained a steam heating element, the mold fitting like a clam shell on top of the hull mold;

(3) A synthetic rubber bladder (about ⅛″ thick with semblance of material appearance and pliability to the heavy inner tube used in automobile tires) cut and formed to the shape of the hull and, in the process, fitted inside the two molds and inflated to push the laminate into position against the female molds; and

(4) Electric light, water, air and steam services and an overhead crane.

As a matter of interest, the one day fabrication of the first Navy plastic boat, the largest monolithic plastic object built to date, is presented in some detail. The faces of the molds were first coated with a parting agent and then with a sprayed film of pigmented resin. Next, successive layers of fibrous glass mat (about 1 yard wide) were laid in the hull mold and each layer successively sprayed with a polyester resin, the amount of resin used with each layer having been predetermined in order that a uniform

resin concentration would result in the laminate. A total of twenty layers of mat were laid in the hull mold; some terminated inside the turn of the bilge, others just beyond the turn of the bilge, the rest extended up above the sheer line far enough so that enough mat remained to form the side deck; resulting in finished hull thicknesses tapering from ½ inch on bottom to 5/16 inch at gunwale.

Prior to installing the last six layers of mat a non-rigid wood assembly, comprised of the keel, engine girders, transverse and bulkhead frames, was installed. This framework was prefabricated over a full size wooden plug. The wooden plug (built to the molded dimensions of the boat by the Taylor Model Basin) has performed yeoman's service, having been used in the manufacture of the one-half inch aluminum plate mold as a bending and fairing form and as a welding jig. It was next used as a form for the manufacture of the rubber pressurizing bladder and lastly as a jig for the manufacture of the wood framing. After the installation of the wood framing, the remaining six layers of mat were laid in the form and impregnated. The entire inner surface of the hull was then covered with polyvinyl alcohol sheeting (this material resembles cellophane except it is much tougher and will elongate over 30% without rupture) and taped in position. At this point the pressurizing bladder was installed, about ½ pound per square inch of air applied and a framework installed therein to support the bladder in the location over which the deck was to be placed. The portion of the badder in way of the decking was also covered with polyvinyl alcohol sheeting. The forward and side decking was then laid in place and impregnated in successive layers, with deck frames sandwiched in at the proper interval to join with the hull frames. The deck mold was then lowered into place and secured with locking pins, caul sheets at the shear line were installed and steam lines connected to the heating elements. Pressure was gradually increased in the bladder to the working pressure, 30 pounds per square inch. Inspections inside the bladder and some "tamping" of the bladder around frames were made to ensure equalization of pressure against the laminates. The deck mold was fitted with an air lock to the bladder. Electric lights and telephones were installed inside the bladder. At this juncture

the molds resembled a small submarine "frozen" in concrete, the air lock being the conning tower. After the bladder was brought up to working pressure, steam was turned on in the heating elements, bringing the temperature of the laminate up to 177°F. in about an hour. Curing at this temperature and 30 pounds pressure proceeded for two hours. Steam was then turned off, the air pressure dropped rapidly to atmospheric, deck mold and bladder removed and the hull lifted from the hull mold.

The hull then presented a clean, smooth and egg-shell finish in Navy gray color. It weighed 2666 pounds. To complete the boat only the following work was necessary:

(1) Installation of bulkheads, engine bearers, walking flats and canopy by gluing and bolting to structure already cast.

(2) Installation of machinery, electrical equipment and fittings.

The results of tests made on samples of plastics of the same materials and identical fabrication procedure as the plastic boat, show the following physical characteristics, which have been tabulated to the right of other construction materials, for comparative purposes: (See table on page 20.)

The pattern of further development in the manufacture of plastic boats depends, to a large degree, on the fabrication of the 28′ plastic boats under the second contract. This contractor will follow the procedure found to be successful in smaller items. This involves:

(1) The laying up of the laminates dry on an inverted male cast aluminum mold;

(2) The covering of these laminates with a female plastic mold (made simply by plastering over the male mold to the thickness of the hull and laying up fiber glass cloth with contact resins to a thickness of about 1½ inches);

(3) Flooding the laminates with resin pumped at from five to ten pounds pressure;

(4) Heat is not required as room temperature casting resins will be used. Obviously, this process precludes the inclusion of the deck structure integral with the hull. To install frames, the first ply of fiber glass cloth will be laid all over the male mold and into the frame recesses, followed by three or four plies of unidirectional fiber glass and several layers of sisal fibers in the

recesses. The succeeding layers (seven in number) will then be laid over the entire assembly.

PROPERTY	*MATERIAL*				
	Wood (Douglas Fir)	*Steel (Medium)*	*Bronze (Mn)*	*Aluminum (For Boatwork)*	*Laminated Plastic Boat*
Flexural Strength, Flatside, PSI	11,200	60,000	55,000-115,000	13,000-42,000	31,100
Flexural Strength, Edgewise, PSI	.....	60,000	55,000-115,000	13,000-42,000	37,300
Flexural Modulus of Elasticity (Flatwise), PSI × 10^5	16.4 (9.8 for plywood)	290	145	100	11.24
Flexural Modulus of Elasticity (Edgewise), PSI × 10^5	.....	290	145	100	13.63
Tensile Strength PSI	10,750	60,000	55,000-115,000	13,000-42,000	27,000
Tensile Modulus of Elasticity PSI × 10^5	.....	290	145	100	17.66
Compression Flatwise, PSI	920	60,000	120,000	.....	44,300
Compression Edgewise, PSI	6,720	60,000	120,000	.....	21,600
Shear/To long edge of sheet, PSI	1,130	45,000	.....	.....	3,220
Shear/To short edge of sheet, PSI	190	45,000	.....	.....	3,340
Izod Impact (Edgewise) $\frac{\text{Ft. Lb.}}{\text{In. Notch}}$	.....	32	.....	.....	26.07
Specific Gravity	0.44 (Dry) (0.59 for plywood)	7.85	7.4-8.9	2.55-2.75	1.64
% Water Absorption	48	0	0	0	0.17
Tensile Strength/Spec. Grav.	24,400	7,650	7,440-12,920	5,100-8,730	16,500
Flexural Strength/Sp. G./Flat	25,450	7,650	7,440-12,920	5,100-8,730	19,000
Flexural Strength/Sp. G./Edge	.....	7,650	7,440-12,920	5,100-8,730	22,700

The program has been implemented as a result of the excellent achievements of the first plastic boat. The boat has been assigned to a small carrier for use and further tests. A contract is about to be awarded for further development of manufacturing processes and the building of two boats for one of the new prototype combatant vessels. Contracts have been awarded for the construction of five 36-ft. plastic craft, each having a different underwater form. As these craft are for an experimental program, the specifications call for a service life of only one year. Each boat

will be built on cheap, "one-shot" molds. One contractor will use the wood framing of his plaster mold for the boat frames, knocking the plaster out after the boat has been cured. Another contract is nearly completed for the manufacture of 67 plastic canopies for plywood boats already completed or in the course of fabrication.

It is desired to impart some of the enthusiasm and stimulate further development in the field of plastic boats. The following are considered thought provoking:

(1) In the field of plastics, tests show that fibrous glass mat and glass cloth as laminates presently have the highest tensile and flexural strength-specific gravity ratio. Current developments in glass promise much stronger glass laminates for the future. However, glass is the most expensive reinforcing material on a pound basis. It may be feasible to sacrifice strength to some degree and sandwich other materials. It has been found that the use of cotton duck as an outer skin laminate provides better resistance against abrasion than glass laminates.

(2) Resin content is approximately 50 per cent, by weight, of a finished laminated structure. Resins with quick or delayed thermo-setting properties are available through usage of different catalysts. Pigmentation of resins is an accomplished fact. Flame-resistant resins are at hand. (Resins used in the first boat will support combustion to a limited degree.)

(3) Cast and plate aluminum molds for the 28-ft. boat were difficult to make and were expensive, but they will last indefinitely for mass production. The one-shot molds for the 36-ft. craft represent the other extreme. Perhaps, future permanent molds will be built of medium steel, or of concrete with a smooth, non-cracking (possibly plastic) surface veneer.

(4) The question of laying up of laminates, whether on an inverted male mold or in a female mold remains unsettled.

(5) Exhaustive tests reveal that pressures used to date in fabrication, 10 pounds per square inch with the vacuum bag method to 100 pounds per square inch with other methods, produce excellent results if the pressure is applied uniformly. Each manufacturing procedure, the vacuum bag, the pressure bag or bladder and the matching die, presently has strong proponents.

Nevertheless, starved resin spots or excessive resin deposits are by-products of excessive local pressure differentials. It is believed that the use of hydraulic in lieu of pneumatic pressure will be a step forward. Further, air or water could be preheated.

(6) The building of small boats has been more of an art than a science. So our thinking is predicated on wood construction. The method of framing the first plastic boat is an illustration. Framing or reinforcement and foundations must be adapted to the new material (laminated plastic) and to new manufacturing processes. Further, plastic boats lend themselves to curvatures not readily attainable in conventional boat construction.

(7) The possibility of impregnating the outer skin of plastic hulls with an antifouling preparation should be undertaken in spite of the ease with which marine growths can be removed from a plastic boat.

(8) Advantage should be taken of the dialectric qualities of the Material. Bare electric wires might be inserted in the laminates during the laying-up process.

(9) Cost estimates show approximately 90 per cent savings in hull construction time or about 45 per cent reduction in man-hours for the completed 28 ft. plastic boat when compared with a similar wood boat. But, the cost of the mold must be amortized. In this case, 40 plastic boats must be built to break even cost-wise with plywood. Beyond that number, a savings slightly in excess of 40 per cent can be expected on each plastic boat built. At the other extreme, the 36 ft. plasic craft, together with their cheap molds cost only half that of similar craft of wood. Mold costs can be reduced. Substantial savings in man-hours of fabrication will come with the development of more production-like techniques. Further savings in costs and essential manpower can be made because 90 per cent of the labor required in the construction of plastic boats can be semi-skilled, about 50 per cent can be female.

(10) The extension of plastic construction to larger boats or small vessels. It is my opinion that at this time craft up to 110 feet in length (sub-chasers) could be produced efficiently, economically and expeditiously on a mass production basis.

It is believed that the entire subject and particularly the last paragraph presents a challenge to naval architects.

FREE PISTON GAS GENERATOR BRIGHTENS GAS TURBINE FUTURE.

By Commodore Lisle F. Small, U.S.N. (Ret.)

Paper read before DIESEL SYMPOSIUM of Diesel Engine Manufacturers Association held at Cornell University September 3, 1948.

The Author.

Commodore Small graduated from the Naval Academy in 1917 and during World War I served in the Atlantic. He graduated from the Submarine School, New London, Connecticut in 1920 and began a long attachment to submarine activities including a command. He completed a post graduate course at Columbia University (M.S. in 1925) majoring in Internal Combustion Engines. In 1931 he was selected for Engineering Duty Only and began a career which included duty in the Bureau of Engineering and three cruises in Navy Yards. During World War II he served as Assistant Head of the Shipbuilding Division of the Bureau of Ships. He retired in 1946 from command of the Norfolk Naval Shipyard to become President of United Engineering Company, San Francisco, California. In the Spring of 1948 he joined Lima-Hamilton Corporation. Hamilton, Ohio, as Director of Research.

Engineers everywhere are becoming aware that a new era is dawning in which the gas turbine appears to be destined to play a role of ever-increasing importance. It is proper therefore to keep on inquiring into the probable nature of this new way of producing power so as to be able to evaluate correctly its possible impact upon the existing methods and be prepared to profit from its development. At any rate, this is my excuse for presuming upon your patience with my analysis of the situation, which led me to the choice I have made after a lifelong association with power production in the U. S. Navy. Your Navy is by its very nature and traditions ever alert to any promise of a better propelling plant to realize greater speed or greater endurance.

Basically, every gas turbine plant must consist of only three elements—a means for precompressing air, a combustion chamber

for burning fuel in that air, and a turbine wheel to expand the resulting gases of combustion and convert their heat energy into the mechanical energy of a rotating shaft to be absorbed by a suitable load. However, usually that is about as correct as stating that a steam plant consists of a boiler and a turbine. Does it not seem terribly unfair to designate the rest of the machinery in a modern steam plant as mere auxiliaries? Actually, that is the very part of the plant which kept steam competitive and alive through its long career of usefulness.

Nevertheless a gas turbine of the original classical simplicity was actually built in 1940 to drive an emergency electric generator installed in a bombproof room in Neuchatel, Switzerland. It performs satisfactorily, but its thermal efficiency is low, 16-18%, and therefore there are auxiliaries galore in the conventional gas turbine plant today. I wonder if any of you have seen or read about the Navy's experimental gas turbine unit at Annapolis? Well, one must be charitable lest someone might wonder why the heft of some men is being carried around just to keep their brain alive, if brain be the only thing of value in a man.

Most discussions about the subject of gas turbines begin with the airing of apologies as to why they were so long in coming and why it requires a turbine four times as big as the pay load it can produce. Most of the power developed by the conventional gas turbine goes for compression of air, only a small part of which is needed for combustion. The rest is required to dilute the products of combustion in order to reduce their temperature below the limit at which the turbine blading material can handle these gases.

Turbines and compressors are manifestly subject to inevitable losses common to all machinery. It is easy to see therefore that, given a mass flow of hot gases theoretically capable of producing 100 HP., the turbine must actually produce not less than 80 HP. in order to provide 60 HP. for air compression, which theoretically requires only 48 HP., to be able to carry a 20 HP. useful load, assuming an 80% efficiency for turbine and compressor. If, on the other hand, only a 69% efficiency is realizable on these same units, then the turbine would produce only 69 HP., while the 48 HP. for compression of air would require 69.6 HP. and

therefore no excess for carrying a useful load could be produced. Should one be tempted to reduce the size of the compressor to increase residual power then the gases will be too hot to be handled by the turbine blades because of lack of sufficient air dilution.

As one would suspect, the original simplicity of the basic gas turbine plant subsequently has been violated by the addition of improvements in the form of such auxiliaries as heat exchangers or regenerators, intercoolers and two-stage compression, additional combustors for reheating of gases between two separate turbines, etc. Every new addition or modification has created a new "cycle" and theoretical men delight in discussing new cycles. I do not want to be understood as criticizing the "thinking man," and I am well aware, too, that the very walls of this renowned institution of learning might wrinkle in disapproval of any irreverent reference to professors and learning. It only occurred to me that while it is true that "East is East and West is West and the twain shall never meet," yet on the other hand it took centuries of speculation in erudite Latin and Greek upon the roundness of the earth, and not until it was made understandable to the mind of a sailor did Columbus act upon it and supply us with a proof.

As a Naval officer, I have been for years on this dividing line between East and West, between science and sailorman. I have recently made my choice and have now embarked upon my own voyage of discovery to prove that fifty years of experience with Diesel engines (encompassing most of my own engineering experience) is a much more likely basis for a future gas turbine plant, at least for applications sensitive to high efficiency as well as light weight and small bulk which definitely includes the locomotive and the Naval services.

What is it then that the Diesel engine has taught us? I am convinced that what we can evolve from fair consideration will contribute to the success of the gas turbine plant in the form of a new concept of a reciprocator-turbine combination.

The Diesel engine came into being as a compression-ignition engine and therefore developed combustion in high pressure air as distinct from combustion in atmospheric air under a boiler or at moderate pressures in an Otto cycle internal combustion engine.

The combustion space of a Diesel engine cylinder, being about 7% of the piston displacement, is usually a very flat pancake over the crown of the piston, as compared with the enormous furnace volume of a boiler, a sizeable space in a gasoline engine or the substantial size of the combustor of the conventional gas turbine. The Diesel engine has proven that it is possible to burn fuel in a compact combustion space if it be done in high pressure air.

The high-speed Diesel engines of high specific output used in the Navy's propelling plants require 2.5-5.0 cubic inches of piston displacement per BHP. releasing from 22,000 to 44,000* BTU. of the heat of fuel burned per hour for every cubic inch of their combustion spaces, or 38 to 76 million BTU. per hour per cubic foot. Compare this with 250,000 to 600,000 BTU. per hour per cubic foot of boiler furnaces for service and experimental Naval boilers and up to 2.5-5.0 million BTU. per hour per cubic foot claimed for combustors of conventional and experimental gas turbines.

Despite such high heat releases the combustion chamber of a Diesel engine has been developed to such perfection that there is no maintenance problem associated with parts comprising it in a successful design of engine. In turbo-jets of modern airplanes like in all conventional gas turbines, periodic replacement of the combustor is accepted as inevitable in much the same fashion as re-bricking of a boiler furnace.

The secret of this highly successful performance of the Diesel combustion chamber lies in its *intermittent operation, a basic and most important difference,* as compared with the continuous operation of the conventional gas turbine combustors and boiler furnaces. Such combustion lasts but a few degrees of crank angle and is accompanied by cooling of the combustion chamber walls first gradually by expansion of the products of combustion and then more violently by discharging them into the atmosphere and finally by scouring and scavenging the cylinder with cool charging air for the succeeding cycle.

Furthermore, the Diesel engine represents the most efficient way of producing power by combustion, which means that a

*Assuming fuel consumption of 0.4 lb/BHP/hr equal to heat release of .4 x 19,000 = 7,600/BTU/BHP/hr for 19,000 BTU/lb fuel. Assuming further that combustion is completed within the 7% clearance volume equal to .07 x 2.5 = .175 cu. in./BHP producing a heat release of 7,600/.175 = 44,000 BTU/hr per cubic inch.

greater part of the heat content of the fuel is actually converted into useful mechanical energy and does not stray into the walls of the structure to plague the operators.

At the very beginning Rudolf Diesel had to compromise with practical engineers and surrender his original idea that the "rational heat motor" required that all heat losses be eliminated as much as possible and therefore the walls of the cylinder and combustion space should be insulated and not cooled. The success of the development of the Diesel engine rests undoubtedly upon the fact that water-cooling of cylinder walls was introduced on the first engine to operate beyond the single explosion of the original experiment. Water-cooling is responsible for the excellent reliability and long life of parts exposed to heat and the low loss of efficiency caused by it is well worth the price. The combustors of the conventional gas turbine on the other hand follow the boiler practices in relying upon refractory metals and frequent replacement.

Shortly after the World War I a French engineer (by adoption), Pescara, applied for a patent for an internal-combustion motor compressor unit "in which the transformation of energy into work is effected without any kinematic connection of the piston intended to transform a reciprocating movement into a rotary movement" (from U.S. Patent No. 1,615,133). He must have been influenced like many another future engineer standing on a railroad platform and wistfully listening to the unusual "sobbing" of the locomotive's Westinghouse compressor for air brakes—the most likely progenitor of the Pescara free-piston compressor. And why should the reciprocating movement of the piston be transformed into rotation of a shaft, anyway? It would seem to me that this transformation and the crank mechanism used for that purpose bear the same relation to the intermittent pressure on the piston as money and the bank bear to everyone's economic effort. Both are essential only because income and outgo are improperly synchronized. As a result, the income is first converted into a bank deposit or it serves to increase the speed of a rotating mass in anticipation of withdrawal to cover an expenditure or to carry the load between two successive energy impulses by slowing down. In a four-cycle engine, for instance,

there is one working stroke for every four strokes of the cycle and in a single-cylinder unit that would mean one impulse every two revolutions of the shaft. Without the mass of a heavy flywheel to accelerate and decelerate, the RPM of the engine shaft would vary excessively.

The crankshaft offers also a convenient means of holding together and synchronizing the impulses of individual cylinders in a multi-cylinder engine. There the impulses are closer together and the need for heavy rotating masses is reduced correspondingly.

An air compressor piston with fuel combustion on one side of the piston and compression of air on the other manifestly has very little need for energy storage during the cycle and in a single cylinder form there is no synchronizing problem. Thus the piston can be made free to accelerate by combustion impulses and decelerate by the compression of air in the compressor cylinder on the way out, then accelerate again by the residual energy in the clearance space of the compressor and decelerate by the compression of the air charge in the working Diesel cylinder on the way in. For perfect balance of reciprocating masses, two pistons moving in opposite directions are used, and to make the compressor suitable for a variety of pressures to suit the energy developed by combustion, these pistons will have to be stepped. All of the air compressed by the larger pistons may be permitted to pass through the ports on one end of the working cylinder after the ports on the other end have let out most of the hot gases of combustion, thereby thoroughly scavenging the cylinder and cooling the walls of the combustion chamber in the center of it. This will result in a hot mixture of air and products of combustion which is then ready to be expanded in a normal gas turbine. Such a unit then becomes a Free Piston Gas Generator. We are searching for a better name because of the possible confusion that arises over the use of the word "generator" which, by squatter's rights, belongs to the electrical fraternity.

The Free Piston Gas Generator, as we are developing it, is fundamentally a single cylinder unit in which two opposed pistons operate as balanced pendulums of infinite radius with fuel burned between them every time they come together as in an opposed piston Diesel engine for the specific purpose of generating gas at

a high temperature and pressure suitable for use in a gas turbine. The real heat cycle is not completed until the generated gas is expanded in the gas turbine. In other words, the Free Piston Gas Generator assumes all the functions of the steam boiler and its

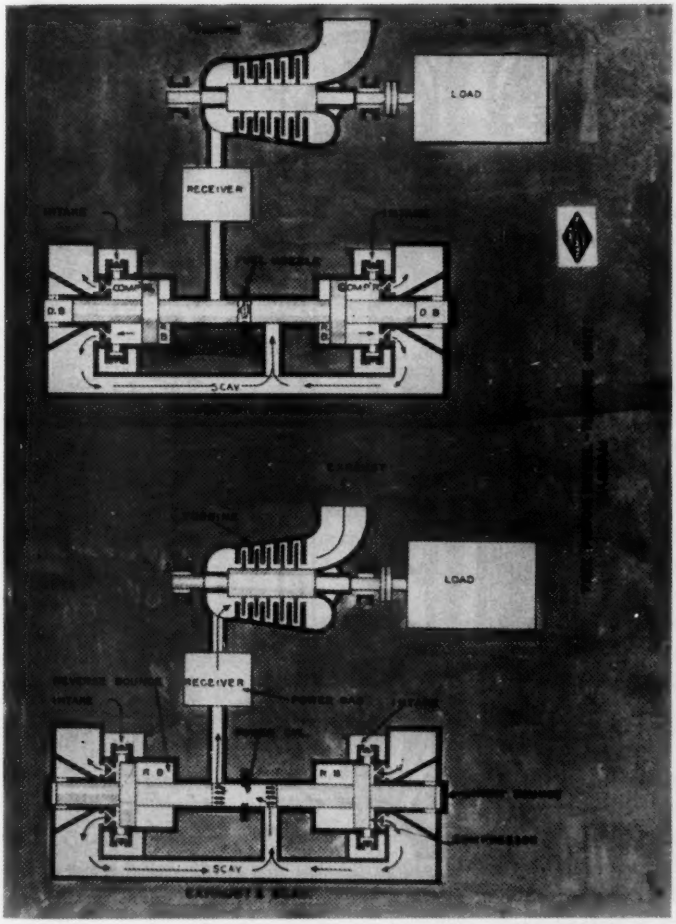


FIGURE 1.

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a high temperature and pressure suitable for use in a gas turbine. The real heat cycle is not completed until the generated gas is expanded in the gas turbine. In other words, the Free Piston Gas Generator assumes all the functions of the steam boiler and its

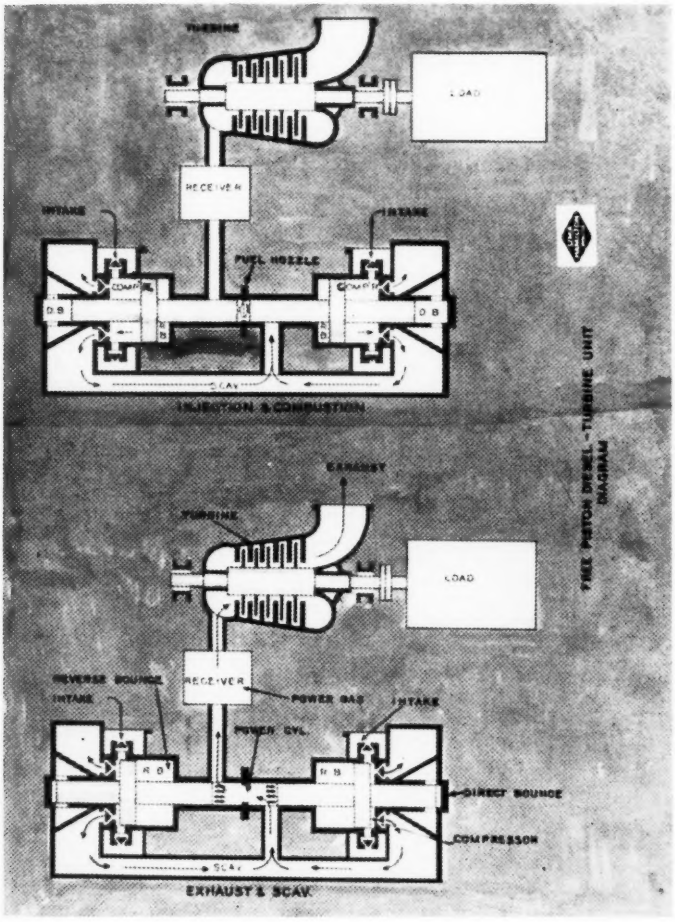

FIGURE 1.

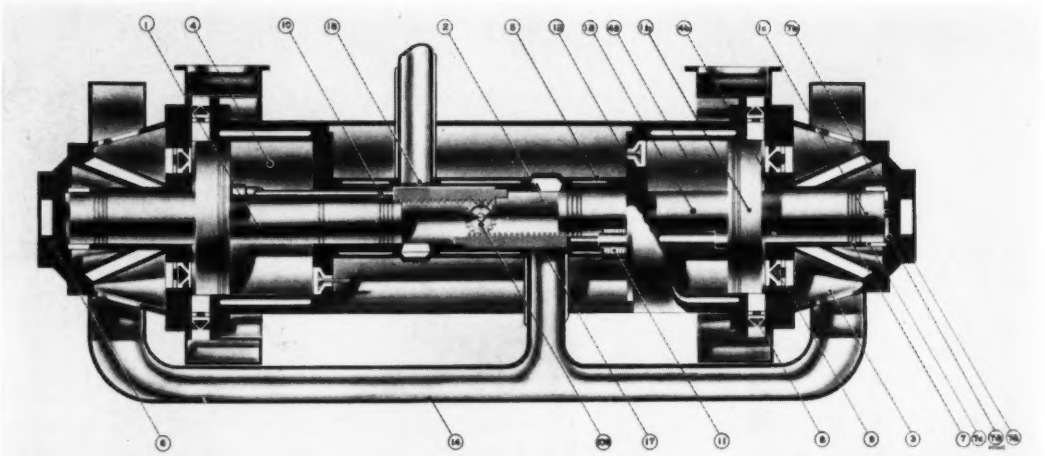
1 PISTON ASSEMBLY
1a POWER PISTON
1b COMPRESSOR PIST ON
1c BOUNCE PISTON
2 POWER CYLINDER
3 SCAVENGE AIR RECEIVER
4 COMPRESSOR CYLINDER
4a REVERSE BOUNCE SPACE
4b COMPRESSOR SPACE
5 POWER CYL. LINER
6 DIRECT BOUNCE HEAD
7 DIRECT BOUNCE SPACE
8 COMPRESSOR INLET VALVES
9 COMPRESSOR DISCH. VALVES
10 SYNCHRONZING RACK
10a SYNCHRONIZING PINION
11 STUFFING BOX
12 DUMP VALVE
14 SCAVENGE AIR DUCT
17 INLET PORTS
18 EXHAUST PORTS
SCHEMATIC DIAGRAM OF A FREE PISTON GAS GENERATOR

FIGURE 2.

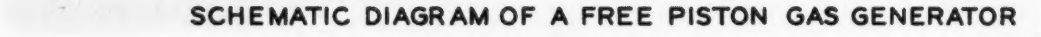

FIGURE 2.

multiplicity of auxiliary equipment, but it is by mechanical design very much like a Diesel engine and no one need be apologetic for copying the many good points of a Diesel engine. I for one have devoted most of my adult life to the development of Diesel engines and I love them. If this new development did not use the wonderful characteristics of a Diesel engine for burning oil efficiently, I doubt if I would have attached my interests to it. As our national economic society is organized today, *oil is life*, and it behooves us to ultilize it with the greatest economy that engineers can provide.

Figure 1 shows schematically a gas generator connected to a gas turbine with the pistons in both of their extreme running positions, while Figure 2 gives an enlarged diagrammatic section of the free piston unit itself. The power cylinder is very much like that of an opposed piston Diesel engine. It has open ends, ports adjacent to its ends, and fuel injection nozzles on its center radial plane.

Relatively large diameter compressor pistons are on the outer ends of the power pistons. The face of these pistons away from the power pistons compresses and delivers air for scavenging the power cylinder. All air pumped by the two compressors passes through the power cylinder. The space adjacent to the power piston is designated as a reverse bounce chamber used primarily for governing the unit. The outermost end of each piston operates in the direct bounce space whose primary purpose is to store sufficient energy to return the piston assembly to a proper compression position after the space between the two power pistons is properly scavenged. The synchronizing racks are attached to the compressor pistons, and the synchronizing pinion is located on the frame of the unit. The principal duty of this synchronizing gear is to drive and time the fuel injection pump.

There are three principal operating variables. By varying the amount of fuel injected and/or by varying the mass of the air entrapped in either the Reverse Bounce cylinders or the Direct Bounce cylinders, the frequency of reciprocation, the stroke of the pistons and consequently the gas horsepower delivery of the unit can be varied. The flexibility of handling these variables is astounding and gives the Free Piston Gas Generator tremendous

advantages in the art of burning fuel for power generation at varying loads.

Briefly summarized then the Free Piston Gas Generator inherits all the advantages of the Diesel engine and in addition it has no bearings which represent the restraint upon the free motion of the piston, while the force of the combustion impulses is tempered by the elastic cushioning of air compression without developing any piston thrust due to angularity of the connecting rod, which is a real cause of piston ring trouble. However, after questioning of the usefulness of the crankshaft to the solution of the Diesel-compressor problem, I must urge patience and warn against shallow over-enthusiasm. It took many a year after the original Kitty Hawk experiment before the airplane was in a position to challenge the passenger-carrying supremacy of the ocean greyhound. With respect to the Free Piston Gas Generator, I do not mean to infer however that its "Kitty Hawk day" happened only recently.

It is true that hundreds of free-piston Diesel-compressors have been built for the German Navy by Junkers and the French production of Pescara Diesel-compressors, interrupted by the war, has been resumed on a considerable scale. It is also true, however, that Sulzer Brothers built a 7000 HP. free piston gas generator and gas turbine plant suitable for merchant ship propulsion, thoroughly tested it and found it unable to compete commercially at the time with a more conventional gas turbine plant operating on a "semi-closed" cycle.* It is again true that Pratt & Whitney have spent much money and effort developing a free piston gas turbine plant for airplanes for the Navy Bureau of Aeronautics and came to the conclusion that it was a "hard nut to crack" in altitudes over 20,000 feet and switched over to developing other means, more promising of immediate utilization. The British are also reported† not to be making much progress with the development of free piston gas generators under Pescara license.

However, the company with which I am now associated has been testing a Free Piston Gas Generator unit for nearly three years for the Bureau of ships and the results obtained have

*Some Researches in Internal Combustion Engines by James Calderwood, JOURNAL A. S. NAV. ENG: Aug. 1946 p. 493.

†Marine Gas Turbines—Abstract of original British paper in Mech. Eng'g June 1948 p. 537.

justified a considerable program of development for the Bureau, and the Company, on its own initiative, is now building a full-sized locomotive power plant of this type for one of the oldest and proudest railroad systems of the country. We have carefully analyzed by known facts the reasons given and implied about these reported failures. We feel that the failures were due to conditions which do not apply to the type of plant we are developing, which is based primarily upon the highly successful submarine engine practice of the U. S. Navy. It is not necessary for it to be competitive with other types of gas turbine plants which fail to qualify for locomotive and Naval applications on the ground of weight and bulkiness or low efficiency. Neither is it to be subjected to the more stringent specifications of the aircraft industry, which at the moment is under the magic spell of jet propulsion. I feel, however, that the unquenchable fuel thirst of the turbo-jet and turbo-prop airplane propelling plants is not likely to be improved so radically that they will become comparable with the free-piston reciprocator-turbine combination as we intend to apply it.

I am aware, of course, that several locomotives with the conventional gas turbine plant are now being built in this country and I wish them sincerely the best of luck, "but as for me and my house," as The Good Book puts it, we are ready to work hard for the realization of the indisputable potentialities of the free-piston reciprocator-turbine combination.

The principal need in any development is to get together a group of smart and enthusiastic, but practical and hard-boiled men determined to prove that they are right, back them with ample funds and facilities and then join the cheering section on the sidelines to be ready with encouraging noise whenever the need for it appears. We sincerely believe that we have just such a gang—devoted, consecrated and tough. We aim to develop and to perfect the most efficient way of producing power in a light, compact, reliable and simple combination of reciprocator and turbine, free of any bearing troubles or unbalanced mass effects of the crank-shaft engines, and requiring no bulky heat exchangers to be cleaned and kept tight or burned-out combustors to be replaced. We thoroughly believe that our contribution will be a national asset.

I have said very little about my ideas concerning the future possibilities of the Free Piston Gas Generator. I could not drive myself to be specific as we all know engineers recoil from speculation. In this connection I would like to close with a little personal history to illustrate my point.

When a young Naval officer, I made my decision to specialize in internal combustion engines. At that time I could have gotten very large betting odds that never within my lifetime would Diesel engines become a major factor in Naval craft propulsion except in submarines. In October, 1944, the total amount of installed horsepower of Diesel engines for propulsion in your Navy passed that of "Mighty Steam." At that time I still had a reasonable life expectancy of tweny-five years. How soon I will see history repeating a similar rapid step of progress, I hesitate to forecast, but I am hopeful all of us will be pleasantly surprised soon. But omitting all speculation, forgetting all dreaming and all phases of mental activity related in any way to either of these, I offer to you this little parcel of personal philosophy:

"Life is not worth living unless one is willing
To burn as brightly as convictions will permit."

MAGNESIUM AS A SHIPBOARD MATERIAL NEGATIVE CONSIDERATIONS.

By Commander T. W. McConville, U.S.N.R.

The Author.

Commander Thomas Williams McConville, U.S.N.R. Reported for active duty in 1939. Was assigned to damage control duties in U.S.S. *Texas* in 1941. He has completed courses in Chemical Warfare, Naval Damage Control and Naval Fire Fighting. He served for five years in large combatant ships as Assistant Damage Control Officer and as the Damage Control Officer. His experience as Damage Control Officer in ships included enemy action in which his ship sustained bomb and shell fire damage with subsequent responsibility for controlling both fire and flooding, and repair of damage.

At the Cessation of hostilities he was assigned to duties in the Bureau of Ships and is presently in charge of the Damage Control desk.

Historically the entry of the United States into World War II and subsequent experiences demanded that our ships, of combatant and auxiliary types, be provided with additional quantities and more varied types of offensive weapons and defensive devices. These additional guns, directors, radio, radar and other types of offensive and defensive gear were installed in ships in commission and were built into ships under construction.

It was soon realized that these new improvements were being installed at the cost of reducing the stability of the ship. In some cases a positive stability factor was reduced to practically negative, and as such these ships basic characteristic to withstand damage was so critical that concern was evidenced as to their ability to perform the duties assigned.

Directives were issued by the Navy Department that weight compensation for every pound installed in ships must be made whenever practicable. In many instances this demand for weight compensatoin required the removal from the ship of many items which were normally considered of importance, or which contributed to the convenience of the men who served in them.

In a broad sense everything which did not contribute to the offensive or defensive characteristic of the ship was eligible for removal should it be required for weight compensation.

Because of the more or less limited amount of non-essential weight in ships, it was necessary to explore the possibilities of reducing weight by using lighter weight metals having the ability to give satisfactory performance comparable with steel and brass. At the present time aluminum is being used to some extent in replacing the heavier metals formerly used.

Paralleling the research program established by the Navy to find ways and means of reducing weight, there is another program directed toward utilizing a metal that is available in unlimited quantities.

These research programs are at present focused upon magnesium and magnesium alloys as they have the characteristic of being much lighter in weight, and available in unlimited quantities.

To recapitulate, the Navy needs a metal lighter in weight than steel or aluminum that is available in large quantities. Magnesium and its alloys have these advantages.

However, let us consider the principal disadvantages of Magnesium and its alloys.

Of secondary importance, magnesium and many of its alloys do not have the ability to resist corrosion from salt water or spray. This characteristic can be overcome, to a more or less degree, by the application of surface coatings. The problems involved in developing correct formulae for such protective coatings are now being studied in Naval research laboratories.

The greatest disadvantage and real hazard of using magnesium and its alloys in ships is their ability to ignite at comparatively low temperatures and the ineffectiveness of the usual shipboard fire-fighting methods and equipment to extinguish fires involving magnesium and its alloys.

Considerable research has been undertaken, by commercial and service agencies, to determine the possible hazards and methods of extinguishing magnesium fires. At the present time nothing has been developed that can be considered as a practical method or technique for extinguishing magnesium fires should they occur in ships.

It is considered appropriate to disclose at this time certain pertinent remarks which are contained in reports made by the research agencies investigating magnesium fire hazards.

It has been reported that magnesium is of special interest because it will burn when molten or in a finely divided form. When molten or burning its affinity for oxygen is so great that it will decompose water and combine with the oxygen in the water. In the same report we find that magnesium must be raised to or near to its melting point before the affinity for oxygen becomes great enough to start immediate combustion. This temperature which is known as the ignition temperature, will vary with the form of the metal and to a slight extent with the composition of the alloy. In the case of magnesium chips or dust as produced by ordinary commercial machining operations, it is about 600° F. to 700° F. The solid metal shapes must be heated to about 1000° F. to 1200° F. to cause combustion. However, a long exposure of the solid metal to a temperature in the range of 800° F. will sometime cause ignition.

One of the numerous Underwriter's tests involved a pile of varied size magnesium alloy castings, gates and risers weighing 250 pounds. The pile was charged with 15 pounds of mixed magnesium turnings and moistened sawdust. A train of moistened magnesium sawdust was laid out for ignition. At one minute and ten seconds after ignition of pile the solid metal started to burn and at two minutes and thirty seconds molten metal was collecting at the base of the fire. Sprinklers were turned on at three minutes and five seconds. Nine sprinklers, each discharging about 25 gallons a minute were used. Spattering was caused. Thermocouples indicated a maximum temperature of 100° F. before sprinklers were turned on. Temperature dropped to 88° F. one minute later, then rose rapidly. At four minutes and 22 seconds explosive puffs started and occurred at frequent intervals until six minutes and five seconds when these explosive puffs reached their maximum intensity. At this point thermocouples indicated a maximum temperature of 795° F. dropping to 115° F. in the next minute. At nineteen minutes and thirty seconds the fire was no longer visible, however, steaming and crackling was present. The fire was determined to be extinguished at thirty-three minutes and thirty-seven seconds and the sprinklers were closed.

Subsequent inspection of this pile indicated that most of the unburned pieces were originally near the top of the pile but as the fire developed quickly the castings and other pieces of solid metal were heated and began to melt and burn, with drops of molten metal falling to the base of the pile. This resulted in centering the fire near the base which facilitated the cooling of the pieces at the top with water from the sprinklers.

It is considered important to note that it required thirty minutes and thirty-two seconds to extinguish a fire involving only two hundred and fifty pounds of metal.

One must consider the hazard to personnel who must fight fires involved with magnesium and its alloys.

Reports indicate, that because of the volume of smoke and intense heat, firefighters are unable to get within an effective working range of the fire. And that water, when applied from hose streams, caused violent reactions, scattering magnesium and endangering the firefighters.

Considering the limited space available in ships the hazard to firefighting personnel is considered to be greatly increased.

There appears to be three different conditions or types of exposure causing magnesium fires:

(*a*) A rapid heat input such as might be encountered when magnesium parts are exposed to large flammable liquid fires, or when there is direct flame impingement on the magnesium.

(*b*) A slow heat input with a maximum ambient temperature slightly above 700° F. Most of the alloys contain low melting eutectics which "weep" to the surface and ignite at temperatures considerably below that reported as the melting point of the alloy itself.

(*c*) Exposure to temperature below 700° F. for long periods of time "ten hours or longer", in an air atmosphere. It has been observed that a gray coating forms which probably consists of white magnesium oxide and orange magnesium nitride. Very little is known of this latter type fire, but it is occasionally encountered during aging and stabilizing treatments where the parts are exposed to temperatures below 700° F. for periods considerably in excess of 10 hours.

The most serious fire to which a magnesium alloy part would

be exposed would be that involving gasoline, oil, or a combination of the two. Being liquids, they are capable of flowing and thus give a fire area of much greater extent and intensity than solid materials. Furthermore, gasoline is easily ignited, not only by direct contact with a spark or flame, but also from hot surfaces. There is no way of estimating the probability of magnesium alloy parts being exposed to gasoline-oil fires. The temperature produced by burning gasoline and oil is approximately 2000° F. which is nearly double the melting point of magnesium and aluminum alloys. These temperatures are reached almost immediately and depending upon the size and specific surface of the magnesium alloy part its ignition will require anywhere from one to two minutes to as long as fifteen to twenty minutes.

Magnesium is naturally less inflammable in mass than in more finely divided form. Experiments are reported which demonstrate the difficulty of igniting a block of magnesium by means of a blow torch and at the same time show the ease of igniting powdered or ribbon magnesium. In the case of a large solid mass, with such great heat conductivity, the heat is conducted to all parts so rapidly that the entire mass must become heated to temperature before ignition. On the other hand, if a tip of a welding torch, an acetylene torch preferably, is applied continuously to one spot, a well of liquid metal forms, although the entire mass raises considerably in overall temperature, indicating that the amount of heat input is greater than the conductivity.

The melting points of magnesium alloys reported in trade publications are considerably higher than the ignition temperature. Eutectics and inter-metallic compounds present in magnesium-aluminum-zinc systems begin melting at temperatures as low as 800° F. Small globules of molten metal appear on the surface where they immediately burn. In most applications, including aircraft, composition of the standard alloys has little or no significance from a fire hazard standpoint. It is of importance only in such operations as heat-treating, where certain alloys are held at temperatures close to the eutectic melting point for 10-20 hours.

Preliminary tests performed by Factory Mutual Research Corporation indicate that when parts are maintained at temperatures slightly lower than even the melting point of the lowest melting

compound present, a form of spontaneous heating can take place which will eventually result in ignition. Considering the type of exposure that may be encountered in shipboard fires, it is considered that this phenomena is significant.

Tests have been conducted using aircraft landing mats constructed of commercially available magnesium alloys. It was determined from these tests that such alloys had two inherent deficiencies which were detrimental to their use for this purpose. It was found possible to ignite a runway surfaced with magnesium landing mats which would support its own combustion simply by lighting a small gasoline spill.

Claims have been made that the addition of beryllium in magnesium or its alloys would eliminate, or greatly reduce the fire hazard contingent with their use. Tests have been conducted which indicate that when .005 per cent of beryllium was present, the magnesium alloy could be melted and heated to about 1400°F. to 1450°F. without an undesirable degree of burning, it was also determined that the protective effect of beryllium decreased with the melt holding time, and that subsequent preparation of two melts in graphite crucibles gave strong evidence that iron was reducing the effectiveness of beryllium in inhibiting burning. It is conceivable that under conflagration conditions, a ship would contain sufficient heated iron and steel to overcome the beryllium content of magnesium alloy fittings.

As far as can be determined, all tests to determine the flammability of magnesium alloys have been conducted in open areas on concrete or wooden flooring. No tests are recorded where the ignition of magnesium alloys took place on steel decking. However, demonstrations of the effect of burning magnesium on steel plates have been witnessed by the writer. These demonstrations clearly indicate that steel decking offers no protection and is easily penetrated by molten magnesium, which action in ships would destroy watertight and fire boundaries and endanger all compartments, or tanks below or adjacent to a compartment containing a magnesium fire.

Based on the conclusions reached in these various unbiased tests which in every case indicate the inherent ability of magnesium and its alloys to burn at relatively low temperatures, it is

believed that instances where fire has occurred in Naval vessels, and the resulting heat damage should be presented.

In a certain aircraft carrier the sprinkler heads, manufactured of cast brass, exposed to the fire for 5 to 15 minutes before water was turned on, melted. Others broke off when the water was turned on.

Another case is reported where an area roughly 16′ x 16′ on the starboard side of the centerline sagged two inches. This area was subjected to the hottest part of the fire above and below deck.

Records cite another incident in an aircraft carrier where there was no initial damage to the after elevator, but the intense heat of the fire caused the platform and heavy supporting frames to buckle and fold downward. The heavy plungers buckled and distorted, as if they were constructed of light metal.

Several reports indicate that aluminum valve wheels at hangar deck sprinkler control valves melted.

Planes on the hangar deck burned until all gasoline was consumed and the planes themselves melted down, leaving only the engine.

One ship reported the spread of fire to spaces on the second and third decks was very rapid. Some blasts entered on open hatches and burning gasoline poured down this hatch to the second deck and thence through another hatch to the third deck.

On the basis of observations of damage in ships, in which severe fires have occurred, it is believed that temperatures ranging from 1800° F. to 2500° F. are not uncommon. While these temperatures, in many cases were largely confined to the main and upper decks, there are also a number of cases in which it is believed temperatures exceeding 1800° F. were encountered below the main deck.

From these and other reports it is apparent that temperatures generated by severe fires in ships will ignite all magnesium alloys on which test data is available.

With the facts as presented, it can be established that a fire involving magnesium and its alloys would very probably be present should such metals be installed in ships suffering similar conflagrations.

The previous several paragraphs have been concerned with

the fact that magnesium and its known alloys are inflammable. The second unfavorable characteristic is that this metal, once it becomes ignited is difficult, if not impossible, to extinguish. This characteristic is of greater concern as it affects the entire concept of fire fighting facilities and procedures as now established in the Navy.

Large magnesium fires cannot be extinguished!

Once burning has been established, extinguishment is exceedingly difficult, if not impossible, with any of the extinguishing compounds now known.

The common type extinguishers are ineffective on magnesium fires because violent reactions can be extremely dangerous. Water is decomposed by burning magnesium into its component parts, oxygen and hydrogen, the latter becoming involved in the combustion and furnishing additional heat. Foam is likewise without extinguishing value, as it consists principally of water. Carbon tetrachloride reacts violently with burning magnesium while proprietary dry chemical extinguishers are either decomposed or are inert. Carbon dioxide has been reported as increasing the burning rate to an explosive stage, however available test data does not bear out this statement. Carbon dioxide is decomposed but does not have any pronounced effect on the combustion of magnesium. Specially compounded dry powder as well as cast iron borings, talc, sand and similar inert materials, can smother or confine fires in small quantities of magnesium where close approach for application is possible.

A number of so-called "extinguishers" have been tested using materials inert to magnesium, but which in themselves are flammable. These have not been used on fires involving more than a pailful of scrap turnings. Lubricating oil or mixtures of carbon tetrachloride and lubricating oil are representative of this class and have not proven satisfactory, since it is difficult to apply the mixture to the burning alloy without causing the oil itself to ignite.

Water from sprinklers falling on burning magnesium caused some scattering of sparks or globules of burning magnesium in every test. In several of the tests, it is reported that the discharge of water from sprinklers on burning magnesium produced minor

explosions of a degree of intensity which might be termed explosive puffs. It appears that most of the explosive puffs were hydrogen-air explosions resulting from hydrogen given off by the chemical reaction between hot magnesium and water. While no violent explosions have been reported in any of the tests, it is considered possible to have a violent explosion where considerable quantities of molten magnesium and a mass of water are brought in contact.

It has also been reported that a fire in nearby combustible materials, burning too slowly to operate automatic sprinklers, may develop enough heat to ignite thin-section magnesium, producing a well started magnesium fire before the operation of the automatic sprinklers.

Considering all the tests conducted by using a large variety of materials and equipment it is believed that water sprinkler systems provide the best means of combatting magnesium fires. However, what becomes of the saving in weight by the use of magnesium, if it is necessary to install sprinkler equipment in practically every compartment of the ship?

Continuing research possibly will, at some future date, develop a compound or technique that can be used succesfully in extinguishing fires involving magnesium or its alloys in ships. These are two conditions which should be given thorough consideration in the development of any such material or method: first, the application of any such new development should not necessitate the installation of weight in the ship that would offset the weight advantage afforded by using magnesium or its alloys; second, any compound, either liquid, vaporous, or solid that will decompose, in contact with heat, into combustible or toxic vapors cannot be considered acceptable for shipboard use. As an example, carbon tetrachloride will, when heated decompose with a residual product of chlorine. This fact caused the removal of this compound, for fire fighting purposes, from all ships of the Navy.

With the information available it is believed that any program to introduce magnesium or magnesium alloys into ships should be pursued with caution. And although fully aware and concurring in the necessity for weight reduction, and use of less strategic metals in ships, it is not believed that these urgent needs can, or

should, be resolved by using a material which is known to have hazardous characteristics.

In conclusion, the use of magnesium or magnesium alloys, as now known, should only be considered for use in ships on the basis of need for weight saving against the safety of the ship.

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DIESEL ENGINE IN UNITED STATES NAVY.

By E. C. Magdeburger.

(To the 50th anniversary of the landing of the first Diesel engine in America)

"Only he deserves to live
who does not fear to die"—
Theodore Roosevelt.

"Teddy" Roosevelt is the patron saint of the United States Navy and his birthday on October 27th is celebrated annually as the Navy Day. If there is any one thing which could account for that, it is undoubtedly the pride he took in the creation of the "Great White Fleet," which he gave the whole world a chance to see. "In my own judgement," wrote he in his autobiography, "the most important service that I rendered to peace was the voyage of the battle fleet round the world." This fleet left Hampton Roads on 16 December 1907 and returned to Hampton Roads on 22 February 1909. It represented probably the highest peak of development for the reciprocating steam engine inasmuch as the last battleship propelled by reciprocators, the famous *Oklahoma,* was laid down in October 1912 and was commissioned May 1916.

This briefly was the stage upon which the Diesel engine was destined to enter and in the U. S. Navy is without the slightest doubt where it played its greatest role in World War II.

The Author.

Mr. E. C. Magdeburger was born and received his technical education in old Russia. He served a brief apprenticeship in Germany with a builder of gas engines, worked on Naval construction at Vickers' in England and began his Diesel engine career with a Russian licensee of MAN. He came to this country in 1910. After building large gas engines for Allis-Chalmers and crankcase compression engines for Fairbanks Morse he joined in 1912 Busch-Sulzer of St. Louis, the original builder of Diesel engines in this country, who supplied all engines used in powering Lake submarines for U. S. Navy. Since 1922 he has been with the Bureau of Ships, Navy Department and has frequently contributed to the Journal in the past.

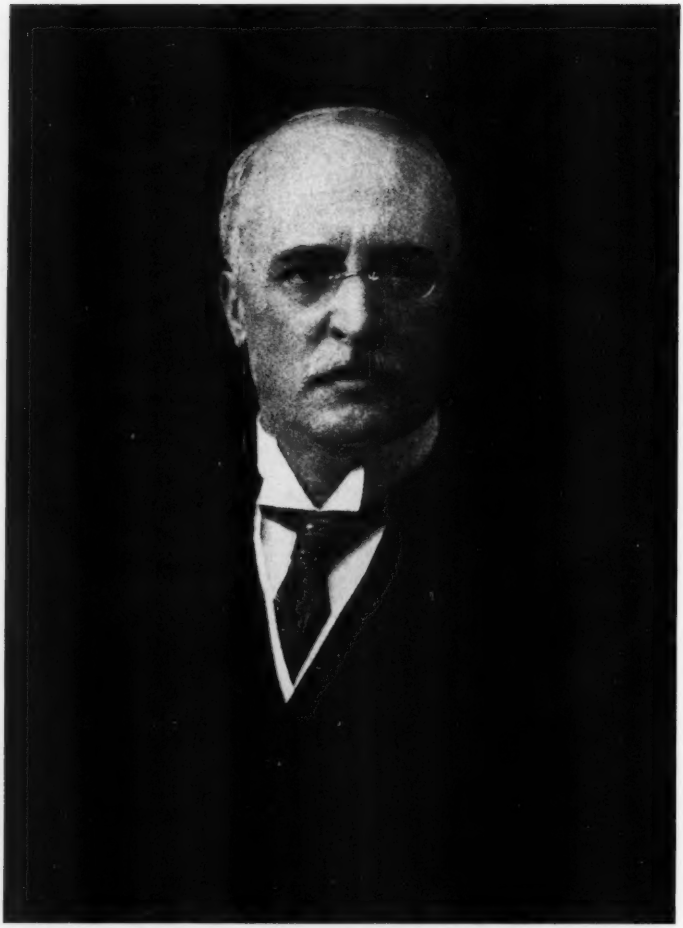

FIGURE 1.—RUDOLF DIESEL BEFORE 1898.

RUDOLF DIESEL, OF MUNICH, GERMANY.

Honorary Doctor of Engineering and of the Technical Sciences, University of Munich. Director, Verein deutscher Ingenieure. Honorary Member, American Society of Mechanical Engineers. Director and Consulting Engineer, Busch-Sulzer Bros.—Diesel Engine Co., St. Louis. Portrait taken at time he delivered a lecture at U. S. Naval Academy, Annapolis, Maryland, April 26, 1912.

The germ of the Diesel idea laid dormant among the abstract conceptions of the Frenchman Carnot, who in 1824 published his revolutionary principle that "the most efficient engine possible must operate on a reversible cycle," which "ideal cycle" has since become recognized as the basic Second Law of Thermodynamics. Years of discussion and argument have passed. The great attraction of the most efficient engine concept must have intrigued many a young engineer and scientist. It was Rudolf Diesel, a German engineer, who studied thermodynamics under Professor Linde of the ice-machine fame in Munich and later was employed by Linde's refrigeration company in their Paris and Berlin offices, who early in 1893 came out with his now famous treatise on "The Theory and Construction of a Rational Heat Motor to Replace Steam Engines and Internal Combustion Motors."

A miracle then happened—the idea was endorsed by many professors and experts and Diesel was successful in obtaining the financial support of some of the principal European engine builders of that day. In the spring of 1893 the MAN Company constructed the first "rational heat motor." It had an uncooled cylinder as originally conceived by Diesel, but it would not run by itself on its first tests in August 1893. Nevertheless, a director of the MAN considered "the process proven in principle" by its occasional explosions. The original motor was rebuilt several times, but at no time did it perform beyond the occasional explosions and much smoking. Only the second, slightly larger unit with water-cooled cylinder built in 1897 became operative and was later used as a demonstrator for the many foreign engineers and businessmen who were permitted to test it as they wished. Adolphus Busch, the famous St. Louis brewer was visiting in Germany in the summer of 1897 and witnessed one of these demonstrations. He called for his Chief Engineer, Col. E. D. Meier, who tested the original unit at MAN plant in September 1897 and then recommended the purchase of the American patent rights.

Several duplicates of this second motor, modified only by substituting the A-frame were built and one of these arrived in America in the spring of 1898—Figure 2—line 1 of table I. A similar motor was built from the same drawings by the St.

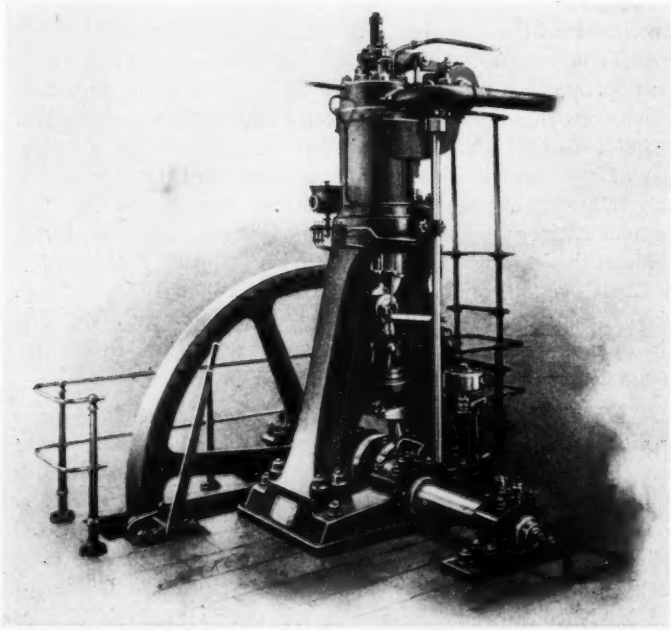

FIGURE 2.—THE FIRST DIESEL ENGINE TO LAND IN AMERICA.

Louis Iron Works to Mr. Busch's order and was installed in his brewery to carry a part of its daily load thus becoming the first Diesel engine in the world to carry a commercial load.

There were many obstacles to be overcome before American business, hypnotized by what appeared to be inexhaustible deposits of crude oil, took serious notice of "the most efficient way of producing power by combustion" and began to adapt it to its own needs. Several starts to produce the new engine in this country had to be made before the firm founded by Mr. Busch in 1912 in cooperation with Sulzer Brothers of Switzerland began to prosper. However, the brewery interests have finally sold it to one of its competitors on December 27, 1946.

The first American design of Diesel engine—Figure 3—already exhibits the important contribution made by America to its

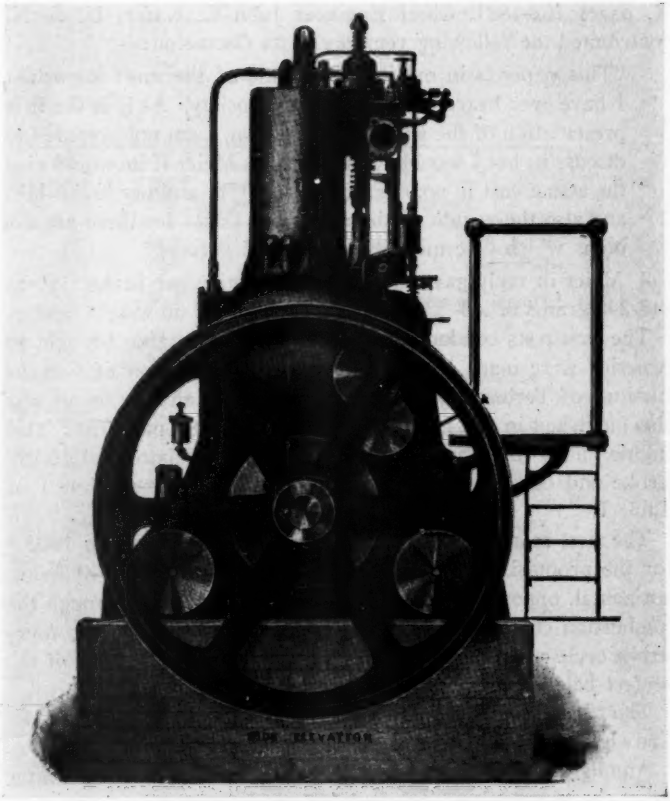

FIGURE 3.—THE FIRST DIESEL ENGINE OF AMERICAN DESIGN.

development—the trunk piston construction, which soon thereafter was adapted by European builders also.

THE EARLY CONTACTS WITH U. S. NAVY.

The first description of the new engine appearing in America was given in a paper by Col. E. D. Meier entitled "The Diesel Motor" and was published in the February 1898 issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS (Vol.

X, pages 162-182). Chief Engineer John C. Kafer, U. S. N. contributed the following remarks to its discussion:—

> "This paper is in my judgment, one of the most important I have ever heard read before any Society. As it is the first presentation of the subject I have seen, I am not prepared to discuss it, but I would like to ask Col. Meier if he would give the actual cost in pounds of oil per IHP. and per brake HP. and also the weight of the motor per IHP., for these are the items which determine its commercial success."

Col. Meier in reply gave the fuel consumption per brake HP. as 218-240 grams or .48-.53 lb/HP/hr, but quoted no weight figures.

The first tests conducted on the first Diesel engine brought to America were made by Professor James E. Denton of Stevens Institute of Technology in June—August 1898 and his report was also published in the JOURNAL (Volume XI-1899 page 73). This engine had a single working cylinder of 10.23″ bore and 16.16″ stroke and developed 20 HP. at about 188 RPM—see line 1 of Table I.

The first marine Diesel engine was built in France in 1902-3 for the propulsion of canal boats. It was of the single-cylinder, horizontal, opposed-piston type with its shaft passing through the combustion chamber in a cooled jacket. It worked on the four-stroke cycle and was capable of "very high speed" because of the perfect balance characterizing its crank mechanism.

The first reversible four-cycle Diesel engine was built by Ludwig Nobel in Russia in 1908.

Among a collection of early pamphlets and advertising matter about the new engine preserved for many years prior to the last war in the technical library of the Bureau of Ships was a letter, a copy of which is being reproduced in its entirety, it being probably one of the first proposals made to U. S. Navy to power any Naval craft with a Diesel engine. However, it is hardly necessary to add that the 50 foot "steam" cutter mentioned therein was never converted to Diesel propulsion, not even years later.

AMERICAN DIESEL ENGINE COMPANY
11 Broadway, New York December 1, 1908
Subject: Two Cycle Diesel Engine for 50 feet Steam Cutters

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TABLE I. DESIGN DATA AND PERFORMANCE CHARACTERI

Line No.	First Built	Builder—Type	Fuel Injection	Starting	Reversing	Cycle	No. of Cylinders	BHP Total	BHP/Cyl.	Bor In.
1	1898	MAN	Air	Air	....	4	1	20	20	10.2
2	1911	NELSECO—VICKERS	"	Non.	Non.	4	4	275	69	12.7
3	1913	NELSECO—MAN	"	Air	Rev.	2	6	450	75	11.0
4	1915	NAVY—MAN	"	"	"	2	6	2500	417	25.2
5	1914	SULZER BROS.	"	"	"	2	6	600	100	12.5
6	1916	BUSCH—SULZER	"	"	"	2	6	600	100	12.5
7	1917	" "	"	Non.	Non.	2	6	300	50	9.5
8	1918	NELSECO	"	"	"	2	6	420	70	10.5
9	1918	"	"	Air	"	4	8	240	30	9.0
10	1919	"	"	"	"	4	6	440	73	13.5
11	1918	BUSCH—SULZER	"	Non.	Non.	4	6	500	83	14.2
12	1919	NELSECO	"	Air	Rev.	4	8	600	75	13.5
13	1919	BUSCH—SULZER	"	"	Non.	2	6	900	150	14.8
14	1918	NAVY—NELSECO	"	"	Rev.	4	8	700	87.5	14.5
15	1920	NELSECO	"	"	"	4	8	1000	125	18.0
16	1922	BUREAU TYPE	"	"	"	4	6	1000	167	17.7
17	1926	" "	"	"	"	4	6	1400	233	20.8
18	1922	BUSCH—SULZER	"	"	"	2	6	2250	375	21.6
19	1933	NORFOLK—BUDA	XLO	Elec.	Gear	4	4	25	6.25	3.6
20	1933	" "	"	"	"	4	6	60	10.0	4.0
21	1936	NORFOLK—LANOVA	"	"	"	4	6	105	17.5	4.2
22	1933	WINTON	Unit	Air	Non.	2	V-12	950	79.2	8.0
23	1935	GM—CDED	"	"	"	2	V-16	1300	81.2	8.0
24	1936	FM CO.	Mech.	"	"	2OP	8	1300	162.4	8.0
25	1936	HOR—MAN	"	"	"	2DA	8	1300	162.4	9.0
26	1941	GM—CDED	Unit	Elec.	Gear	2	8	500	625	6.5
27	1937	ALCO—SULZER	Mech.	Air	Rev.	2DA	7	4200	600	16.5
28	1940	GM—CDED	Unit	"	"	4s	V-16	2000	125	9.5
29	1940	GM—PANCAKE	"	Elec.	Prop.	2	16	1200	75	6.0
30	1941	GM—DETROIT	"	"	Gear	2	6	225	37.5	4.2
31	1939	GM—ELECTROMOTIVE	"	"	"	2	V-12	900	75	8.5

NCE CHARACTERISTICS—DIESEL ENGINES IN U. S. NAVY.

BHP/ Cyl.	Bore, In.	Stroke, In.	RPM.	Displacement Cu. In.	Piston Speed Fpm	BMEP Psi	Specific Volume Cu. In. per Bhp	Remarks
20	10.23	x 16.16	185	1,330	500	64.3	66.5	First Diesel in U. S. A.
69	12.75	x 13.50	400	6,900	900	78.9	25.1	First in U. S. Navy—Subs E-1, E-2
75	11.00	x 12.25	450	7,000	920	56.4	15.55	Hand K classes, modified for L class
417	25.20	x 39.37	130	117,900	853	64.8	47.1	U. S. S. *Maumee*
100	12.59	x 12.59	450	9,410	944	56.1	15.68	Sub. G-3
100	12.5	x 14.5	375	10,670	906	59.3	17.8	Lake L class submarines
50	9.5	x 12.0	400	5,105	800	58.25	17.01	Lake N class submarines
70	10.5	x 16.0	350	8,315	934	57.1	19.8	Sub. M-1
30	9.0	x 12.5	350	6,362	730	85.4	26.5	H and K classes re-engined
73	13.5	x 14.0	400	12,020	933	72.6	27.3	EB Co. O and R classes
83	14.25	x 14.0	410	13,400	957	72.1	26.8	Lake O and R classes
75	13.50	x 15.0	380	17,190	950	73.0	28.55	EB Co. S-1, S class
150	14.87	x 16.37	350	17,070	955	59.7	18.97	Lake S-2, S class
87.5	14.5	x 16.0	350	21,130	933	74.9	30.2	Navy Yard S-3, S class
125	18.0	x 19.0	375	38,700	1187	54.6	38.7	EB Co. T class
167	17.75	x 16.56	425	24,600	1173	75.8	24.6	Government S class
233	20.87	x 20.87	345	42,900	1200	74.9	30.65	Sub. V-4
375	21.63	x 21.63	310	47,700	1120	60.2	21.2	Lake V class
6.25	3.625	x 4.50	1428	185.8	1060	74.7	7.46	Navy Power Boats
10.0	4.0	x 5.50	1700	414.8	1560	67.4	6.91	
17.5	4.25	x 5.50	1885	468.3	1728	94.2	4.46	
79.2	8.0	x 10.0	720	6,040	1200	86.6	6.36	Original Prototype
81.2	8.0	x 10.0	750	8,042	1250	85.3	6.19	Porpoise class subs.
162.4	8.0	x 10.0	720	8,042	1200	89.0	6.19	Plunger-Pollack subs.
162.4	9.056	x 13.38	700	13,570	1561	54.25	10.4	Pompano submarine
625	6.5	x 7.0	1200	1,858	1400	88.8	3.72	YMS mine sweepers
600	16.5	x 23.0	420	65,400	1610	60.5	15.56	U. S. S. *Maumee* re-engining
125	9.5	x 12.0	900	13,610	1800	128.3	6.8	PC451 and class
75	6.0	x 6.5	1800	2,940	1950	89.8	2.45	SC453 and class
37.5	4.25	x 5.0	2100	425.7	1750	99.6	1.89	Landing craft
75	8.50	x 10.0	750	6,815	1250	71.0	7.57	LST's

Rear Admiral John K. Barton, Engineer in Chief, U.S.N.
Bureau of Steam Engineering, Navy Department
Washington, D. C.
Dear Sir:—

Referring to your favor of October 15th, transmitting blueprint, issue #14989, file No. 7-H-152, sent to us on October 15th, I take pleasure in enclosing our blueprint D-602A, showing a two cycle Diesel engine of the same general construction as furnished for the Italian Navy by Messrs. Sulzer Brothers of Switzerland.

The engine you will note occupies a little more space than the steam engine, i.e. slightly encroaches on the boiler room, but in the boiler room there are only two starting tanks on each side, leaving nearly the whole of the space available for other purposes. The oil storage is not shown, but will occupy less than one third of the space required for coal for the same mileage.

In order to bring the flywheel on the propeller shaft, it has been necessary to raise this shaft a little and give it a slight inclination towards the rear. The total weight, including the tanks, is five and one half tons.

Regarding the employment of Diesel engines for marine purposes abroad, we take pleasure in referring to two articles in Cassier's Magazine for November, mailed to you under separate cover, viz. "Internal Combustion Engines for Marine Purposes," by Sir John I. Thornycroft, and "Submarine Naval Warfare," by Major Constructor G. Laurenti, pages 225 and 241.

Yours very truly,
American Diesel Engine Company
E. D. Meier, Engineer in Chief.

What is a Diesel Engine?

The engine we know today by that name was probably not "invented" by Rudolf Diesel if the patent lawyers had the final word in the matter or the green-eyed envy were permitted to prevail. Neither did Christopher Columbus accomplish the original purposes of his journey—discovery of a short route to

the Indies, yet there is no doubt in the mind of the world that he was robbed of the most fitting way to commemorate his discoveries by not naming the New World that he did discover after him. Neither is there the slightest doubt that the engine we have today is the result of the initial stimulus supplied by Rudolf Diesel and is a worthy memorial to his unwavering faith in his idea, his great persuasiveness and dogged persistence. It is fitting therefore that it bear his name even though hundreds and even thousands of engineers individually and in industrial groups have contributed to its eventual perfection in varying degrees.

The attacks on the validity of Diesel's claims to the invention began in 1893 by several contemporary German engineers and professors, and culminated in a book by J. LÜEDERS, then professor of the Technical Institute at Aachen, which appeared in 1913.(2)* Lüeders argued that Diesel invented the "rational heat motor" based on an uncooled cylinder and once this feature was abandoned his claims to the invention were forfeited. He saw in Diesel's later publications(1)* merely an effort to fit the story to the facts as they developed.

The British engineer Alan E. L. Chorlton in 1931 attempted to establish that the British development of the Diesel engine under license from him in reality rested on the work of Akroyd Stuart.(3) "One does not want in any way to detract from the work of Diesel, but rather to indicate that the earlier work of Akroyd Stuart with its potential development has been somewhat overlooked. Perhaps a comparison can be made by saying that Diesel's proposals were theoretical and Akroyd Stuarts practical. Whilst a pure air charge and automatic ignition was used in both, the actual engines as made and run differed, for Akroyd Stuart really used the constant volume cycle and Diesel the constant pressure . . . The standard slow speed marine engine using air injection is always associated with the name of Diesel, though it may be said to fall within the scope of the text of the first patent of Akroyd Stuart."

The British magazine "Oil Engine" (first appeared in May 1933) is still "dealing with compression-ignition engines," but its American counterpart "Oil Engine Power," established 1923, has

*See list of references.

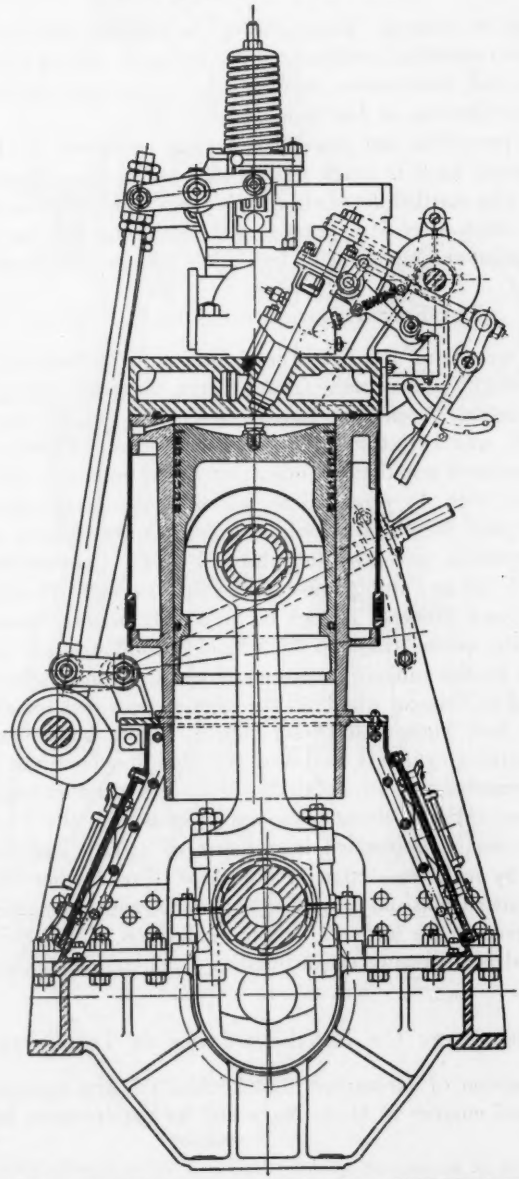

FIGURE 4.—THE FIRST DIESEL ENGINE—FOUR-CYCLE NELSECO-VICKERS FOR U. S. SUBMARINES E-1 AND 2 ALSO F CLASS.

changed its name to "Diesel Power" in January 1930 because of the "overwhelming tendency to use the term 'Diesel engine' for all internal combustion engines with compression-ignition and with introduction of fuel independently of air."

The proverbial last straw that became renowned for breaking the camels back is much less entitled to its fame than Rudolf Diesel who startled the world with his idea and initiated a development, which after fifty years is still continuing and has brought to mankind accomplishments impossible without the Diesel engine.

The First Diesel Engines In U. S. Navy.

The design of the first Diesel engine to be installed in any vessel of U. S. Navy came from Vickers, the great British "Naval Construction Works" at Barrow-in-Furness where James McKechnie was Director of Engine Construction. Vickers had its representative test the first operating Diesel engine in 1897. McKechnie later (4) proposed an internal combustion engine propelling plant for Naval vessels in order to solve the gun arrangement problem, posed by the proposal of Lt. Commander Sims, U. S. N. for an "all big one-calibre ship," through the absence of uptakes and funnels. Though he eventually became renowned as the creator of the "common rail system of solid injection" installed in most British submarines for many years, yet the engine designs supplied to Nelseco, who built the engines, provided for air injection of fuel. These first Diesel engines, Figure 4 and line 2, had four working cylinders, and were installed in submarines E-1 and E-2, commissioned in February 1912. Six cylinder engines of the same cylinder dimensions were installed in the F class of submarines commissioned in summer of 1912. This design is shown by its cross-section of Figure 4. For further details of their construction and of service experience with them, the reader is referred to the author's earlier publication.(12a) For design data and performance characteristics of all engines discussed see Table I.

The Beginning Of The Four-Cycle vs Two-Cycle Feud.

Propulsion of submarines thus became the first application of the Diesel engines in U. S. Navy and its requirements began to

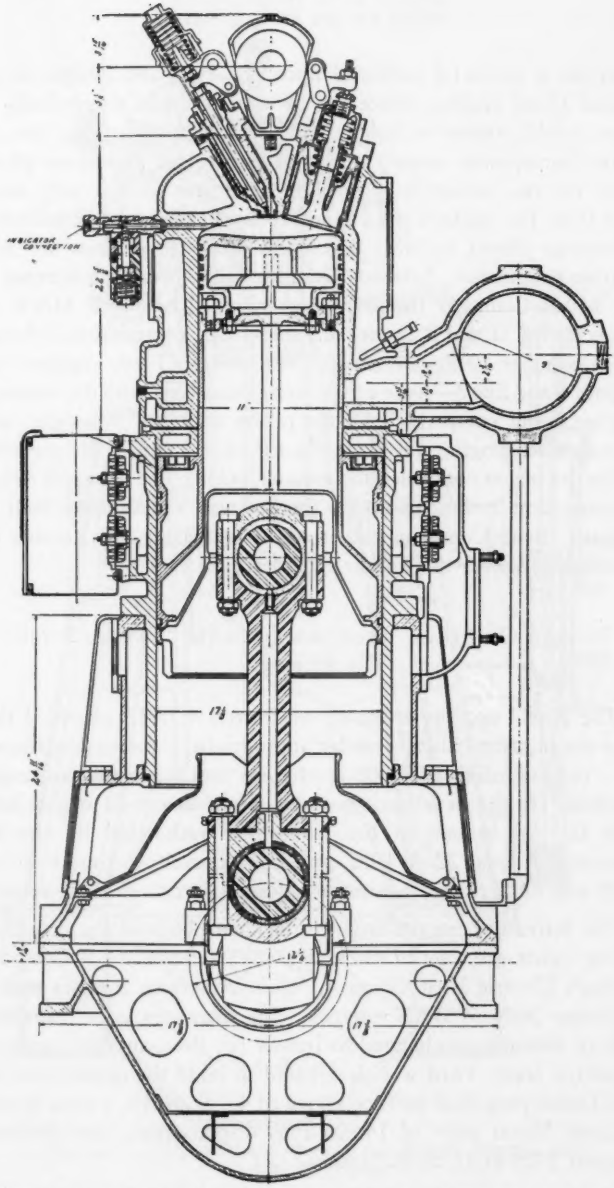

FIGURE 5.—NELSECO-MAN TWO-CYCLE DIESEL ENGINE FOR H, K AND L CLASSES OF SUBMARINES.

exercise a powerful influence upon the type and design of the Naval Diesel engine. Since the two-stroke cycle, theoretically at least, could produce a lighter and more compact engine for any given horsepower output, and since the weight and space allowance for the submarine propelling plant was always very much less than for engines used in other applications, all continental European Diesel builders developed two-cycle engines for submarine propulsion. Nelseco, who meanwhile became a licensee of the MAN Company therefore next supplied two-cycle MAN engines for the H and K classes of submarines, commissioned during the December 1913—December 1914 period. These engines—see Figure 5 and line 3—were of the step-piston type with the scavenging air being compressed by the piston annulus. A single cam-operated scavenging valve was in the closed end of the cylinder while the piston controlled the exhaust ports. The principal design improvement in the engines for the L class of submarines built by Electric Boat Company consisted in increasing the number of scavenging valves to two per cylinder.

Bureau Of Steam Engineering Sends "Diesel Spies" To Europe.

The Naval appropriation act of March 4, 1913 provided that "the unobligated balances under appropriation 'Steam Machinery' . . . not exceeding $250,000 are hereby reappropriated and made available for the development of a type of heavy-oil engine suitable for use in one of the fuel ships authorized by the act approved August 22nd, 1912, and the expenditure thus incurred shall not be a charge against the limit of cost of such vessel."

The Bureau of Steam Engineering then obtained the manufacturing rights and a set of working plans from the MAN Company through Electric Boat Company, their American licensee manufacturing Nelseco-MAN engines. The drawings were translated and all dimensions changed to inches (in thousandths), and the Brooklyn Navy Yard was designated to build the engines for the first Diesel-propelled surface vessel of U. S. Navy, a twin screw, 14 knot Naval oiler of 14,500 tons displacement commissioned October 1916 as U. S. S. *Maumee.*(7)

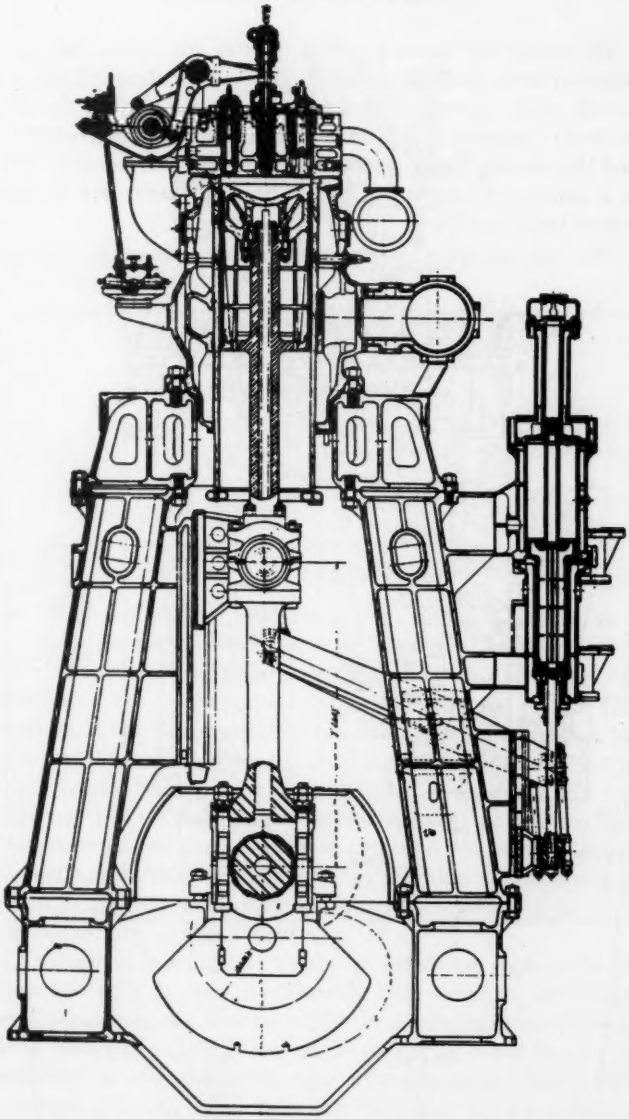

FIGURE 6.—NAVY-MAN TWO-CYCLE DIESEL ENGINE FOR U.S.S. "MAUMEE."

To obtain the "know-how" of the new art of building Diesel engines for the Navy yard, the Bureau sent Lieutenant Chester W. Nimitz with a group of technicians from the yard to the MAN works in Germany just prior to the outbreak of the World War I and the missing finger on the left hand of Fleet Admiral Nimitz is a constant reminder to him of this apprenticeship in Diesel engine building.(5a)

The engines were of the single-acting, two-cycle, crosshead

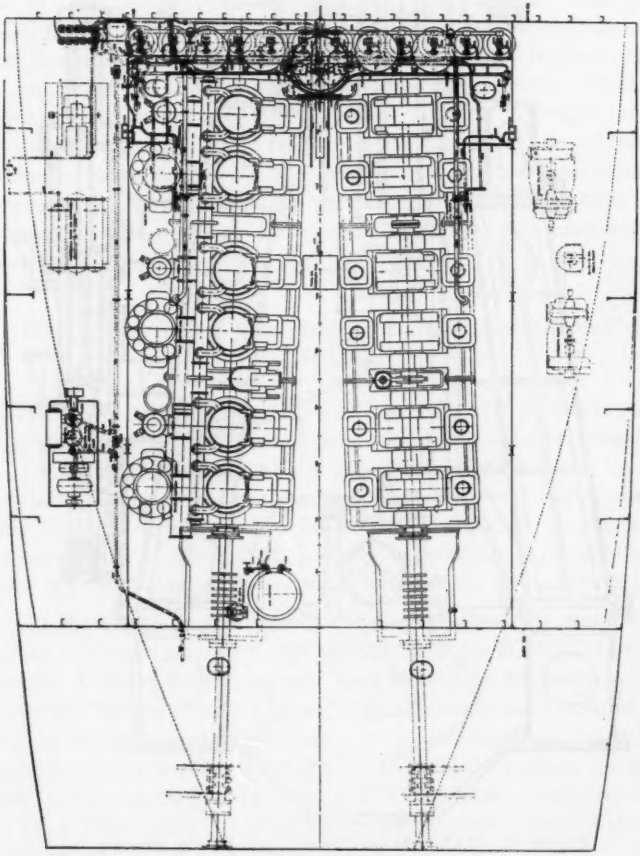

FIGURE 7.—U.S.S. "MAUMEE"—MACHINERY ARRANGEMENT.

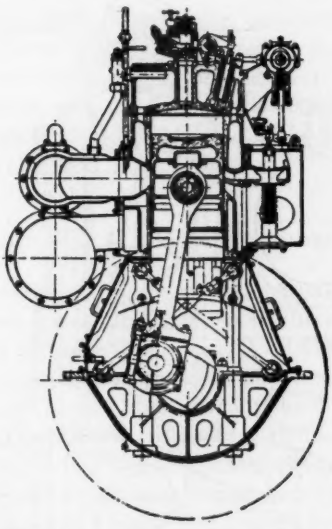

FIGURE 8.—SULZER TWO-CYCLE DIESEL ENGINE FOR SUBMARINE G-3.

type—Figure 6 and line 4—having six working cylinders, three scavenging pumps and three injection air compressors driven by rocker arms, and the usual auxiliary pumps driven from the crankshaft. They were the largest engines built up to that time anywhere in the world and only one other foreign motorship had engines of the same design. U. S. S. *Maumee* did yeoman service during World War I, covering 80,000 miles on numerous trips to Europe, but the hull of this class of tankers proved too light to maintain proper shaft alignment and keep out of trouble with line and crankshaft bearings so that she was decommissioned right after the war. She survived to be re-engined and to render excellent service during World War II.

To secure an independent source of propelling engines for the submarines built by the Lake Torpedo Boat Company of Bridgeport, Connecticut, the Bureau of Steam Engineering purchased a pair of Sulzer two-cycle Diesel engines—Figure 8 and line 5—for installation in the submarine G-3, commissioned March 1915. Lieutenant Felix X. Gygax was therefore sent to Switzerland to act as the Inspector of Machinery during their construction and

FIGURE 7.—U.S.S. "MAUMEE"—MACHINERY ARRANGEMENT.

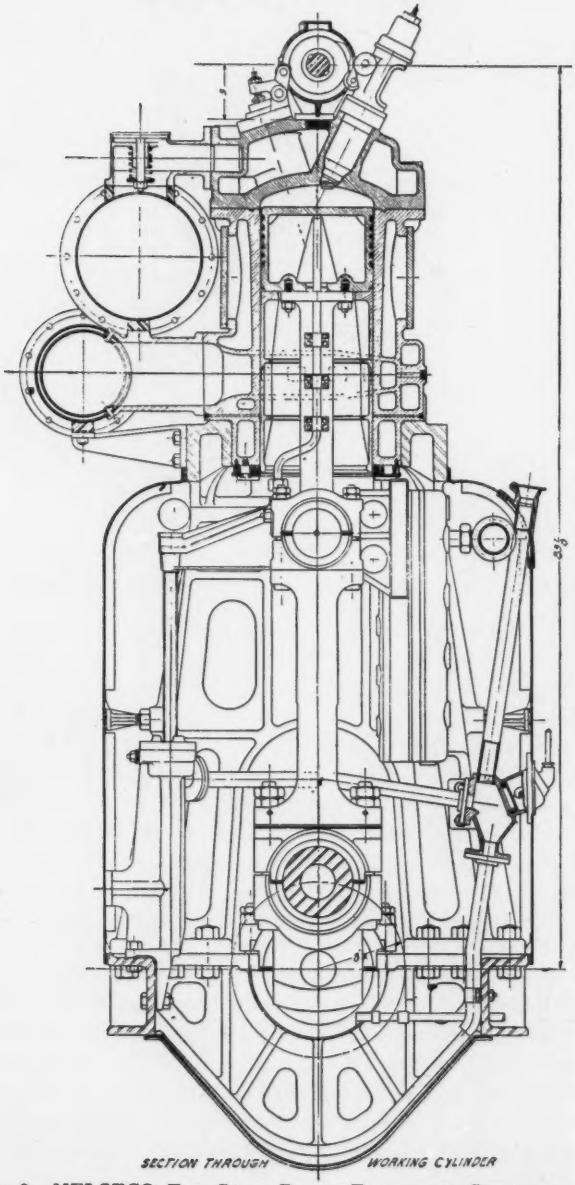

FIGURE 9.—NELSECO TWO-CYCLE DIESEL ENGINE FOR SUBMARINE M-1.

shop tests. As the result of this move, construction of submarine engines of the Sulzer type was organized by Busch-Sulzer of St. Louis during and subsequent to World War I. Rear Admiral Gygax was retired recently following a distinguished career in the Navy as commandant of Norfolk and later of Boston Navy Yard during World War II.

Effect Of World War I On Diesel Engines For U. S. Navy.

With the outbreak of European war, dependence upon foreign designs of Diesel engines for U. S. submarines became a definite handicap. With all communication with MAN severed, Nelseco attempted a re-design of the two-cycle MAN engine and a pair of engines of the new design were installed in the submarine M-1, which was launched April 1915 but commissioned only much later, in February 1918. The step piston was eliminated and two separate double-acting scavenger pumps were substituted—Figure 9 and line 8. Crosshead construction was chosen, which made the engines very heavy, and as the result no duplicate engines of this new design were ever built.

It has been mentioned already that Sulzer engines were imported just prior to the outbreak of war in Europe and Busch-Sulzer Company of St. Louis was commissioned to organize production of submarine engines of this type. The original engines in G-3 was re-designed to suit American shop practices and the requirements of the U. S. Navy and these new engines were used to power the Lake type of L boats—line 6 of Table I. A smaller engine was later designed for the smaller size of submarines—N class—see line 7, used primarily for the training of the submarine crews at the Submarine School at New London, Connecticut.

The dire need for reliable engines for submarine propulsion forced the Bureau to use the 120 HP. 4 cylinder four-cycle ($9'' \times 12\frac{1}{2}''$) engine developed by Nelseco for commercial applications by simply doubling it to produce a light cylinder engine with twin injection air compressors forward—Figure 10 and line 9—developing 240 BHP. at 350 RPM. These were used to re-engine the H and K classes of submarines and though these lost

M-1.

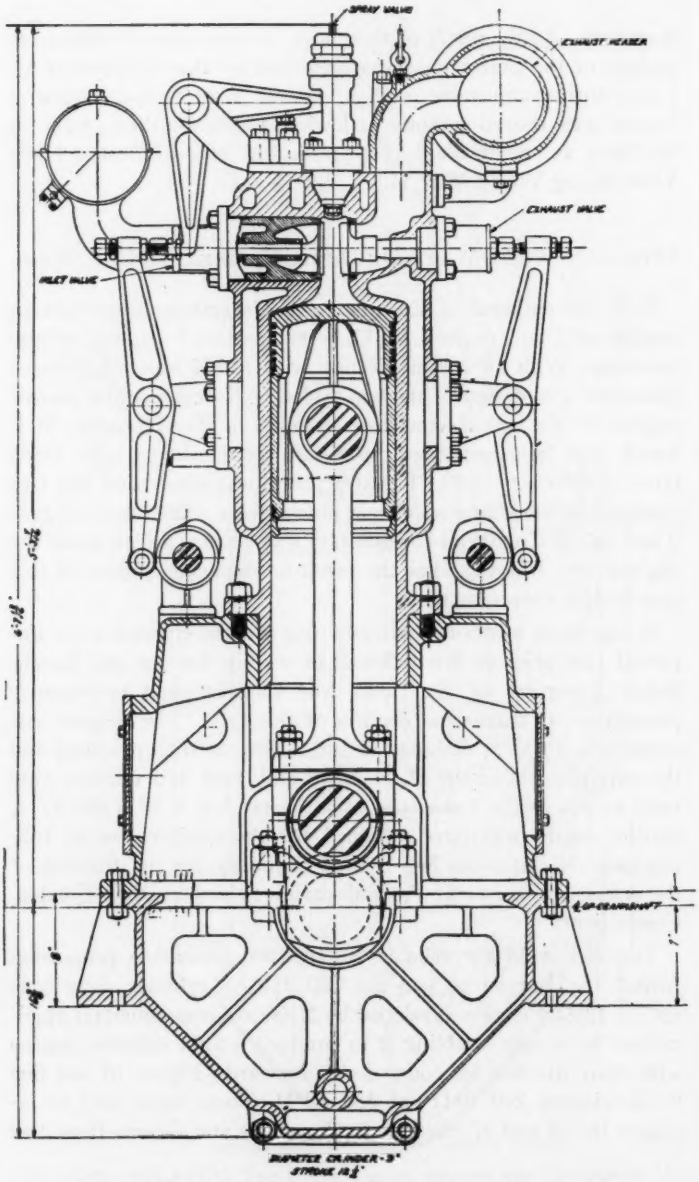

FIGURE 10.—NELSECO FOUR-CYCLE DIESEL ENGINE FOR H AND K CLASSES RE-ENGINING.

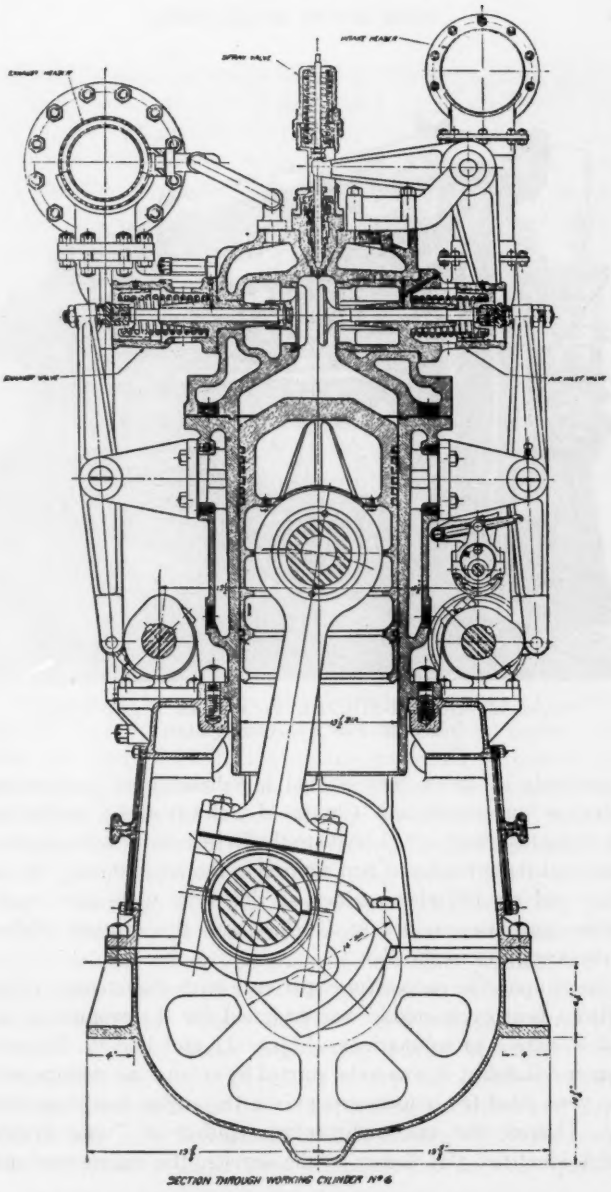

FIGURE 11.—NELSECO FOUR-CYCLE DIESEL ENGINE FOR O, R AND S CLASSES (EB).

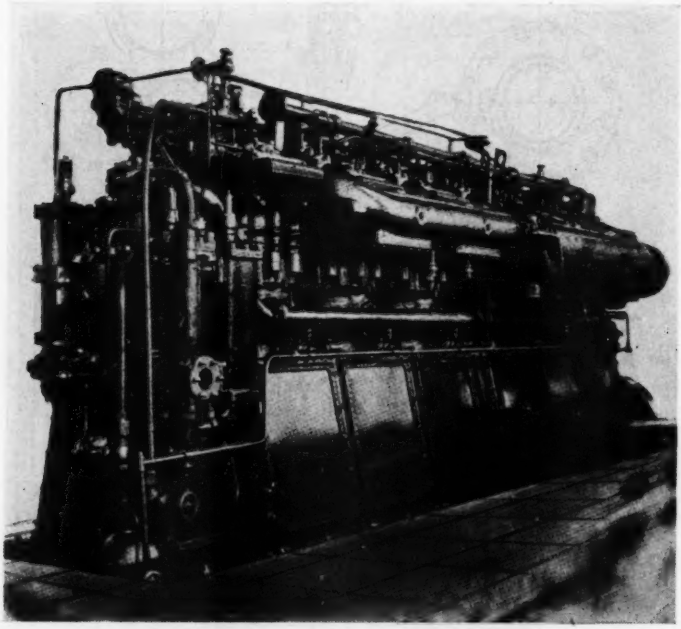

FIGURE 12.—BUSCH-SULZER FOUR-CYCLE DIESEL ENGINE FOR O, R AND S CLASSES (LAKE).

appreciably in speed, they gained in reliability of performance. Later our "super-salesman" Charles M. Schwab sold a sizable number of duplicates of our H-boats to the British with these engines in them and these rendered real service to the allies during the war. Four and six cylinder engines of this type were also used to replace gasoline engines in D class and the original Nelseco-Vickers engines in the E-1.

Based upon the successful experience with this engine, a larger Nelseco four cycle engine was designed for the propulsion of O and R classes of submarines—Figure 11 and line 10. To assure greater reliability, it was to be started by submarine motors, which also were used for maneuvering since the engine was non-reversing. Despite the uncomplimentary epithet of "rock crusher" which identified this design in the service, the engine performed

reliably and helped to generate within the Bureau a preference or prejudice in favor of four-cycle engines which was based primarily on the fact that during the war the Germans used four-cycle engines exclusively. With minor modifications, this same design was later used for a large number of eight cylinder engines —line 12 of Table I—of the EB Co's S class of submarines, which were completed, however, only after the end of the war.

Under pressure from the Bureau, Busch-Sulzer also had to develop a design of a four-cycle engine—Figure 12 and line 11—which were also made non-air-starting and non-reversing and were used for propulsion of Lake type O and R class submarines.

Later several Lake type S boats were also equipped with these engines as a temporary expedient pending production of larger output engines at a later date. These engines did well in the Navy and outlived their hulls. They were then sold and provided with air-starting, and at least one group are in industrial service driving generators in a quarry, as was evidenced by a recent court action in which a Bureau representative was requested to testify.

A Classic "Survival Of The Fittest" Demonstration Planned.

It took a long time before the abstract deductions of Carnot resulted in the bold concept of Diesel's "rational heat motor" and it took the concerted effort of many engineers to convert its occasional explosions into continuous operation. But it took years again before the Diesel engine became an efficient and reliable prime mover even for the easiest practical applications, such as industrial power generation, for instance, where the engine was limited to working cylinders only, i. e. the cooling water and lubricating oil pumps and even the injection air compressor were separately driven; where operation of the engine was limited to one speed and only in one direction of rotation; where the telephone and telegraph were available to procure spare parts for any needed repairs; where engineering advice of experts was easily available in any emergency; where the builders of the engine kept a watchful eye over its operating personnel and were anxious to help with their education and training. However, even reliable performance under these circumstances does not qualify

an engine for marine service and even less for submarine service where only a small group of men are available for engine room duties on the surface and where these same men must respond to the diving signal and assume a part of the responsibility for the lives of all inside the submarine. The submarine engine must therefore be self-contained, driving its own auxiliaries, so that these all stop when the engine is stopped, and start when it is started. It must be capable of operating reliably over a wide range of speeds and in either direction when directly coupled to the propeller shafting, for protracted periods without major repairs and by the skill and the know-how of the small submarine crew.

It therefore soon became evident that it was not the "rational heat motor" part or the Diesel cycle, which plagued the first installations of the submarine Diesel engine (12a)(6) but the auxiliaries driven by the engine, their drives, the leaks of water pumps, the explosions in the crankcases and scavenging air receivers caused by blow-by past piston rings, the inadequate filtering and straining of lubricating oil in circulation, the difficulty of maintaining load distribution between individual cylinders because of leaky packings on fuel injection needles and fuel pump plungers, unreliable air-starting valves, excessive cylinder wear through unsuitable materials used which also contributed to blow-by past piston rings, etc. The first "corrective measures" therefore consisted mostly in compromising by omission of air starting and reversing, in the use of separately driven auxiliaries, in favoring the simplicity of the four-cycle engine, in using small bore cylinders and lower RPM. with consequent increase in specific weight per horsepower.

Thus it was that with the entry of the United States into the European War becoming imminent, construction of three different new types of submarines was authorized on August 29, 1916—S-1 of the Holland type to be propelled by the new Nelseco eight cylinder four-cycle engine—line 12 of Table I, S-2 of the Lake type using a new Busch-Sulzer six cylinder two-cycle engine—line 13 of Table I and Figure 13, and S-3 of so-called Government design built at Portsmouth Navy Yard with Navy-Nelseco engines—line 14 of Table I—of larger bore and stroke but of the

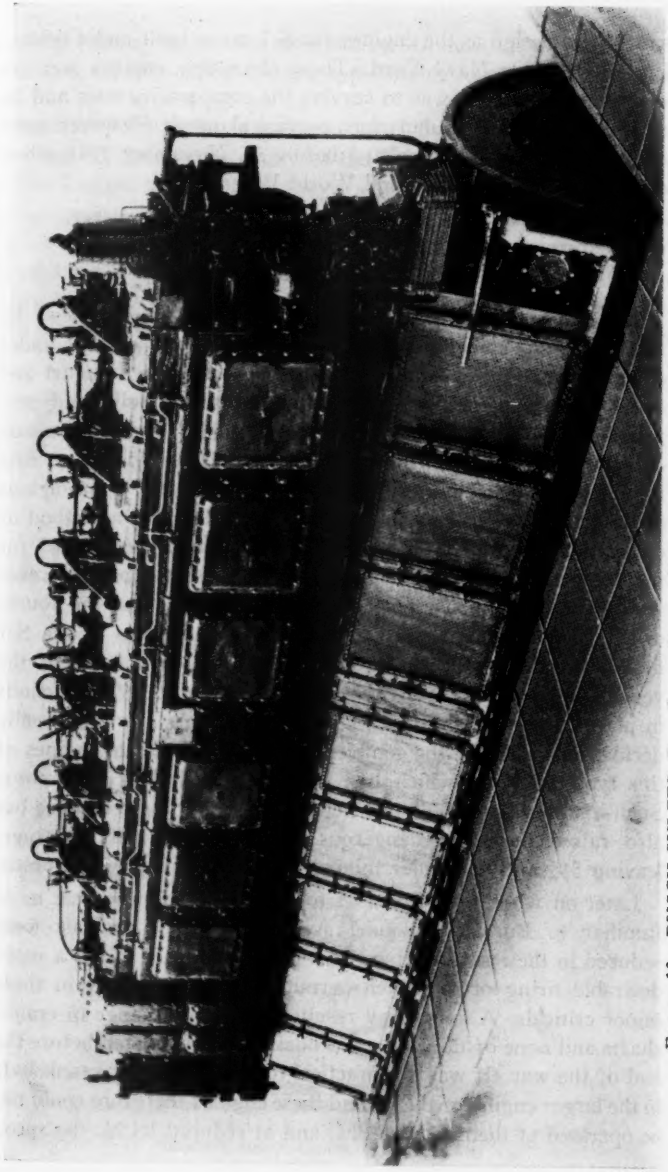

FIGURE 13.—BUSCH-SULZER TWO-CYCLE DIESEL ENGINE FOR S-2 ALSO S CLASS (LAKE) LATER.

same basic design as the engines for S-1, to be built under license by the Brooklyn Navy Yard. These three new engines were to determine which one was to survive the comparative tests and be used in the extensive submarine program planned. However, none of the S boats were completed before 11 November 1918 when armistice was declared to end World War I.

Nelseco S-Boat Engines Taught A Torsional Vibration Lesson.

The Navy had definitely "bit off more than it could chew" by sponsoring the construction of these two different eight cylinder engines, but in the end much was learned to advance the art and one of these engines had been produced successfully and propelled several S boats to glory in World War II. The phenomenon of torsional vibration with its critical speeds had been met first in the Navy Yard built engines for the S-3 and a prodigious amount of effort had to be expended on developing a method of calculation (9) and several designs of friction dampers (10a) for the free end of the crankshaft. None of these dampers however would stand up in service because of the severity of the fourth order critical occurring at rated RPM. The engines of the S-1 first broke their crankshafts only during the final trials of the vessel and since there was room enough to install an 8″ crankshaft in place of the 7″ shaft of the original design, this was finally decided upon despite the completion of many duplicate engines of this type, many of which had already passed their shop tests. Such a stiffening of the shaft not only increased its resistance but also raised the more dangerous major critical out of range, leaving 5½ and 6½ order minors in the upper range of speeds.

Later on when the subject of torsional vibration became more familiar to Bureau personnel even these minor criticals were reduced in their severity by a change of camshaft to get a more desirable firing order which permitted operation even in these minor criticals. A long delay resulted from this change of crankshafts and none of the Holland S boats were completed before the end of the war. It was not practical to install a larger crankshaft in the larger engine of the S3 and these engines therefore could not be operated at their rated RPM. and at reduced RPM. the speed

of the submarine became unattractive. In most of the submarines of this type so-called Bureau type engines were later installed.

Diesel Engines At Beginning Of Post-War Period.

The excellent performance of the German submarines in World War I rested manifestly on the reliable performance of their MAN four-cycle propelling engines which were studied (6)(11) and admired (12) by our Naval personnel and upon conclusion of hostilities were just copied and built in small numbers at the Brooklyn Navy Yard under the "Bureau type" disguise—Figure 14 and line 16. They were of higher horsepower output, lighter

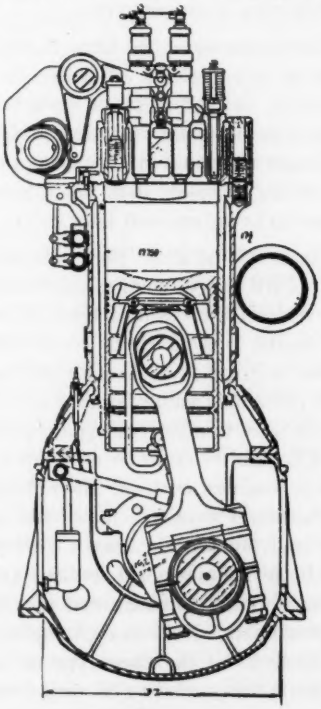

Figure 14.—BUREAU TYPE (MAN) Four-Cycle Diesel Engine for S Class (Government).

weight per horsepower, maneuvered reliably on air and were of simple design based on cast steel bedplates and cylinder housings. Reproduction of these thin-walled and intricate steel castings presented many difficulties. Since only very moderate production was involved, no commercial foundry would undertake it and eventually the Naval Gun Factory at Washington had to supply most of these castings. However, manufacture of these engines by the Brooklyn Navy Yard and production of spare parts for submarine engines of all other designs eventually established a body of competent technical and shop personnel fully capable of coping with the Navy's post-war needs, which were exceedingly light after the war programs were completed.

Eventually a larger engine of the same German design was also reproduced, first in six—line 17 of Table I—and later in ten cylinders. However, the latter never became operative in the German Navy and its torsional vibration problems were not anticipated by its design nor could they be solved by minor modifications of design with the result that the submarines where they were installed had to be re-engined later.

For the propulsion of the first "fleet submarine" (Lake type) authorized July 1, 1918, larger engines were needed and therefore Busch-Sulzer developed a crosshead type of engine—line 18 of Table I. This engine proved to be very reliable, simple and accessible, and was a great favorite with the operating personnel, a compliment of particular value because in addition to the two main engines there were two Bureau type engines driving generators in a forward engine room, to be used for cruising on electric drive. These B-S engines outlived their hulls and eventually helped to solve the power requirements of the Army in constructing the highway and pipeline to Alaska during World War II. Construction of large submarines (Holland type) however had been begun before United States entered the war and postponed in favor of smaller boats. Nelseco eight cylinder submarine engines—line 15, Table I—of the same type as for S boats but of larger size were built but proved to be very unreliable in developing higher powers and speeds. Furthermore the hull design was antiquated so that the submarines as well as the engines were scrapped shortly after completion.

When all of the submarine construction authorized during the war was completed, the submarine strength of U. S. Navy consisted of:

Submarines—first line	50	66,600 SHP.
Submarines—second line	69	56,040 SHP.
Fleet submarines	6	33,000 SHP.
Total	125	155,040 SHP.

Diesel Engines For Battleships.

The enthusiasm for Diesel engines generated by Word War I resulted in diverting some Busch-Sulzer auxiliary Diesel engines similar to Figure 13, line 13 Table I—which were originally intended for fleet submarines, to driving ships generators in battleships *Maryland* and *West Virginia*—(11). However, these installations did not prove successful chiefly because the steam-minded personnel had no patience with the inadvertent errors of these pioneer installations such as connecting the air intake to the ventilating trunk which thus became as a sort-of public-address system the moment the engine was started. Furthermore the engine room compartment was not long enough for the complete unit and the generator had to be connected to the engine by a shaft passing through a bulkhead with the result that critical torsional speeds gave much trouble in the connecting coupling. Furthermore the so-called Rules of Engineering Competition governing operation of engineering plant were not modified to take cognizance of the new Diesel generators and actually made it undesirable to operate them.

Early in 1924, Commander S. M. Robinson, then in charge of Design Division of the Bureau, who had prior to that distinguished himself as the successful advocate of electric drive for battleships (Electric Ship Propulsion, published by Simmons Boardman Company 1922), initiated steps to invite competent Diesel engine manufacturers to submit designs of Diesel engines suitable for re-engining of battleship *Wyoming*. Three engine builders responded and while these preliminary designs were being studied by the Bureau, the Congress on June 6, 1924 passed an amendment to the merchant marine act of 1920, authorizing the U. S. Ship-

ping Board to expend a total of not more than $25,000,000 for converting to Diesel drive steamships owned by the board. Admiral W. S. Benson was placed in charge of this program and Commander Robinson became a member of the advisory committee. The Navy program being in its formative stage it was not deemed wise to push it concurrently with that of the Shipping Board, which was already approved and financed by Congress. Thus the Bureau attempt to develop a Diesel drive for a battleship had to fold up prematurely although efforts to study the possible ways of using Diesel engine for that purpose continued (5b) while the younger submarine officers saw in the tactical advantages of Diesel-propelled surface ships the solution of basic assignments of the Navy (13).

Diesel Engines For Power Boats (12c).

As early as July 1929, two different commercially developed high speed Diesel engines were purchased and installed for a service test in the 50 foot motor launches acting as ferryboats for the Engineering Experiment Station at Annapolis. This service proved to be exceptionally severe and only one engine survived, but it was too heavy for Naval motorboats. In the spring of 1932, the Bureau again canvassed all high speed Diesel manufacturers and accepted for test another engine, but it was also too heavy for ships boats. An important concession had to be made then—the inclusion of a speed-reducing gear in the Diesel engines for Naval motorboats had to be accepted if engines suitable for replacing gasoline engines on size and weight basis were to be developed. With this modification, the specifications issued restricted only the propeller speed to a 600-700 RPM. range for the following three sizes of engines, but not the engine RPM:

Type DA—20-25 HP., weight—750 pounds, overall length—53″

Type DB—60-65 HP., weight—1800 pounds, overall length—78″

Type DC—100-120 HP., weight—2300 pounds, overall length—95″

For the sake of possible reduction of types of spare parts required, they were all to be of the same type of manufacture.

No one manufacturer had all three sizes, but the Buda Company

was eventually prevailed upon to develop the smallest engine and all three were delivered for test at Annapolis before the end of 1933. Meanwhile a "cold room" was installed there to test reliability of starting at an ambient temperature of 20°F and on 3

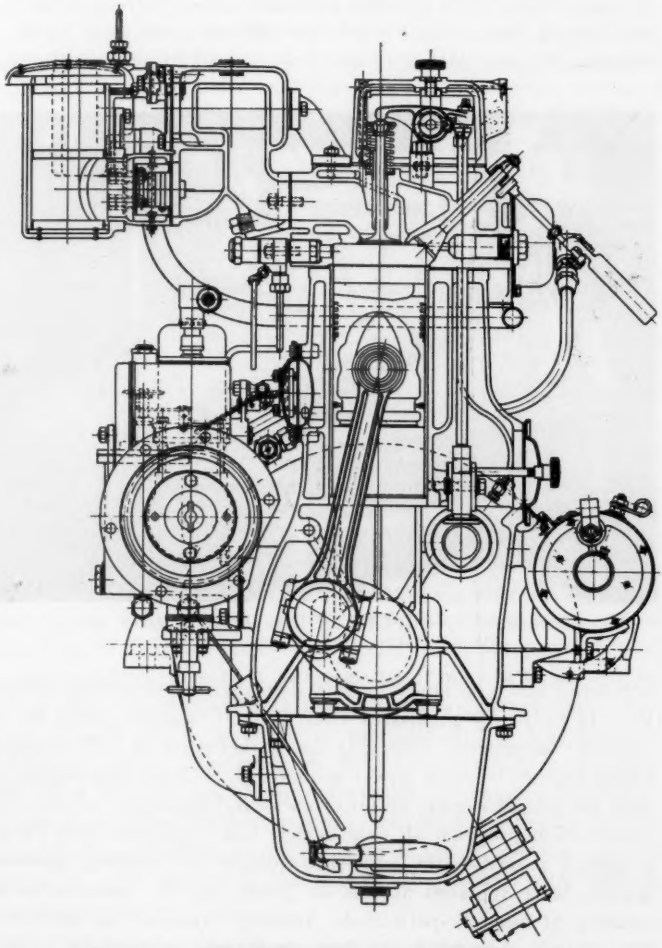

FIGURE 15.—NORFOLK-BUDA DIESEL ENGINE FOR POWER BOATS—CROSS-SECTION.

January 1934 all tests were satisfactorily completed and the Bureau was ready to contract with the Buda Company for the first lot of 158 engines (108 type DA, 22 of DB and 28 of DC) to be paid from NIRA funds (National Industrial Recovery Act of 16 June 1933). This contract included manufacturing rights for the Norfolk Navy Yard where an efficient production of these engines—Figures 15 and 16 and lines 19 and 20 of Table I—was

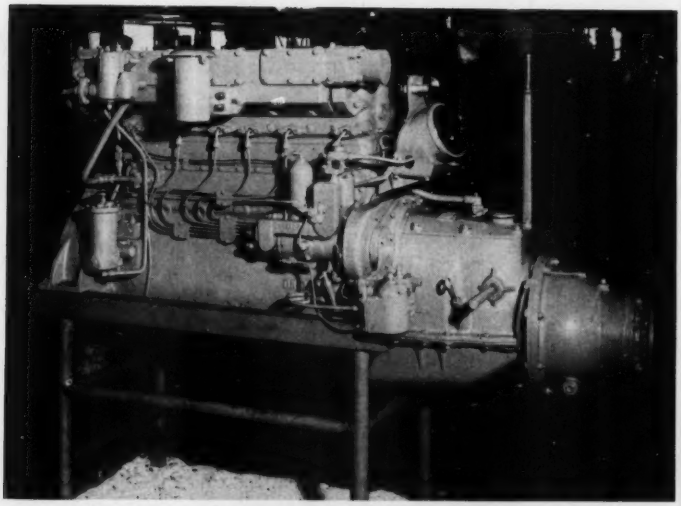

FIGURE 16.—NORFOLK-BUDA DIESEL ENGINE FOR POWER BOATS—LONGITUDINAL SECTION.

organized and continues to this day augmented during World War II by heavy purchases from Buda Company. Thus the ill wind of "depression," following the market crash of 1929, brought Diesel engines to ships power boats of U. S. Navy and beginning with the commissioning of the cruisers and destroyers of the 1933 construction program all new vessels were equipped with Diesel propelled power boats and subsequently all existing gasoline engines were replaced as fast as funds and the manufacturing capacity of Norfolk permitted. Actually, however, no additional DC engines were produced there and these original DC's were replaced by the DD type, which was even lighter than the gasoline

engine it replaced—line 21 of Table I—a DB engine of slightly increased bore, run at higher RPM. and was equipped with a Lanova combustion chamber capable of developing higher BMEP, for which a license had been obtained by the Bureau as well as by the Buda Company.

There were many problems to be overcome, such as development of suitable fuel injection equipment. Here again depression legislation forced the Bureau to develop a new domestic source—the Excello Corporation. Improved designs for the reverse and reduction gears for these units developed by Snow-Nabstedt Company, the elastic mounting of engines on rubber vibration dampers and elastic couplings using rubber ferrules—all contributed to the eventual success of these power boat engines.

New Deal For Diesel Engines In U. S. Navy.

The New Deal for Diesel engines in U. S. Navy began with the appointment of Admiral S. M. Robinson as Chief of Bureau of Engineering, May 29, 1931*. He grasped the opportunity presented by the prevailing depression to employ the idle engineering talent of the nation in creating new types of Diesel engines. Two fundamentals characterized his thinking—the engines had to be much lighter than the existing types and therefore had to operate at higher RPM., and they should not be "tailor-made" to suit each possible application, but be manufactured on productions lines at great savings in cost and effort to suit the nations economic pattern. In March 1932, an invitation was issued to all manufacturers of Diesel engines and to the principal producers of gasoline engines to submit proposals to produce a Diesel engine whose only specification was that it shall weigh not more than 27½ pounds per horsepower, or one-half of the best German submarine engine. By December 1932, five contracts were signed (at about $100,000 each) and eventually five engines varying in output from 320 to 950 horsepower and of widely differing designs (4 cycle supercharged, 2 cycle V, 2 cycle radial, 2 cycle opposed piston, 2 cycle diamond) were built and tested with varying degrees of success (12d).

*After completing four years, he was called again September 30, 1939 and served up to January 31, 1942 as the first Chief of the Bureau of Ships, a consolidation of the Bureaus of Engineering and Construction and Repair effected on June 20, 1940.

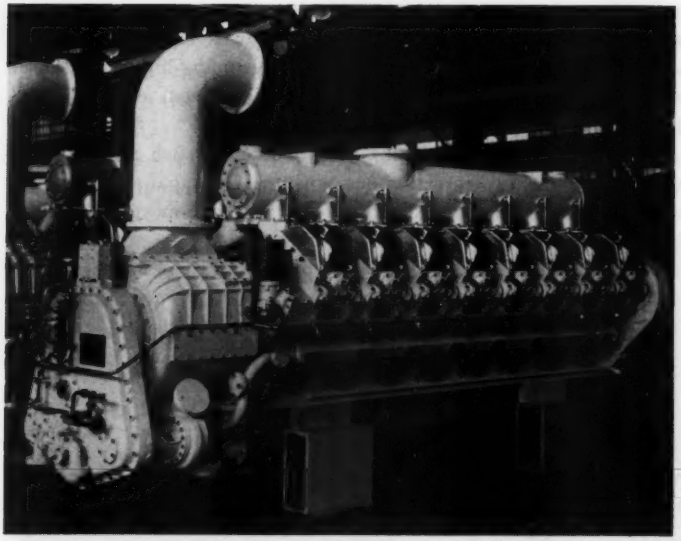

FIGURE 17.—WINTON (GM-CDED) MODEL 201A TWO-CYCLE DIESEL ENGINE FOR "PORPOISE" CLASS.

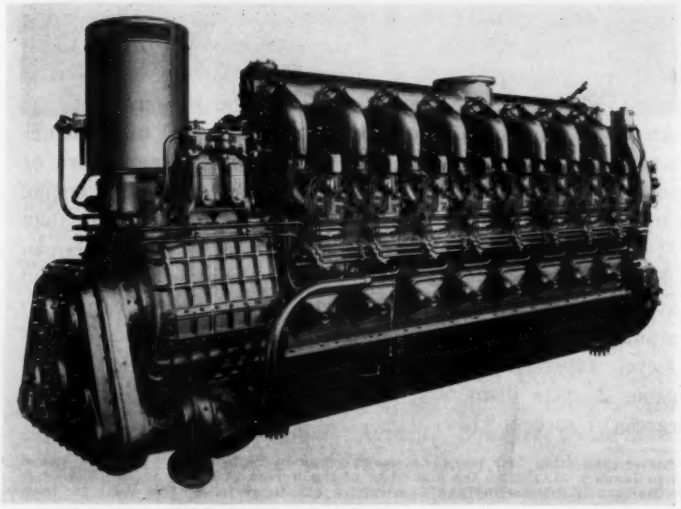

FIGURE 18.—GM-CDED MODEL 278A TWO-CYCLE DIESEL ENGINE.

Only one of these engines—see line 22 of Table I—built by the Winton Engine Company, subsidiary of General Motors Corporation, later re-named Cleveland Diesel Engine Division, survived the severe tests of the development period, primarily because its sponsors were better qualified by experience, had greater resources at their disposal and were content with less radical departure from existing designs and practices. The outstanding features of its design were a welded-steel housing and a unit injector for each cylinder, combining a fuel injection valve with a fuel injection pump plunger and operated by a single cam. As the result of this development, a contract was signed August 3, 1933 for the construction of the first four submarines, *Porpoise, Pike, Shark* and *Tarpon,* employing Diesel-electric drive with four 16 cylinder Winton Model 201A engines driving generators—Figure 17.

This same engine later served as a prototype for two separate developments within the General Motors Corporation, by Cleveland Diesel Engine Division almost exclusively for the needs of U. S. Navy both for submarine and surface ship propulsion—Figure 18, line 23, and by Electromotive Division for the new Diesel-electric locomotives and during the war for many auxiliary and landing craft of U. S. Navy.

The inspiration supplied by the above Navy development program has not failed to produce an important after-effect—two of the Diesel engine builders, Fairbanks, Morse and Company and Hooven-Owens, Rentschler Company, division of General Machinery Corporation, who refused to participate in it originally found themselves forced to develop similar engines at their own expense and requested the Bureau to supervise type-approval tests of these engines in order to qualify them for Naval applications.

Fairbanks, Morse Company developed a two-crankshaft, opposed-piston two-cycle Diesel engine similar to Junkers aircraft type Diesel engine, which was used in large aircraft in Germany—Figure 19 and line 24. It also had a welded steel frame and removable water-cooled cylinder barrels as the General Motor's engine but differed from the later Sulzer D. P. engine by having scavenging ports in the upper cylinder, so that only about 30% of total output was transmitted to the upper crankshaft, which drove the scavenging blower, thus less than one quarter horsepower had to

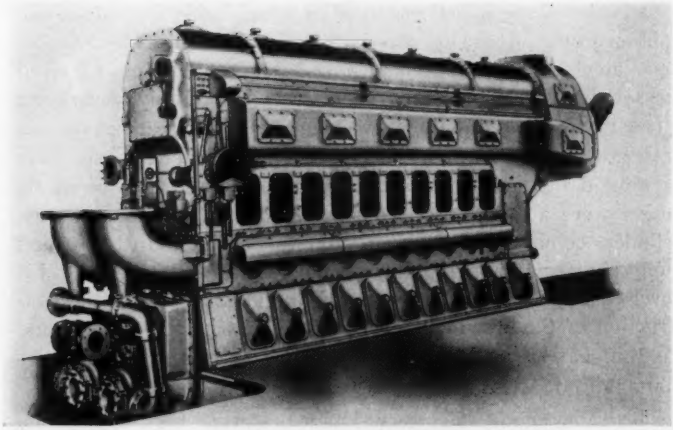

FIGURE 19.—FAIRBANKS, MORSE OPPOSED PISTON TWO-CYCLE DIESEL ENGINE.

be transmitted through the chain or the later developed vertical shaft and its bevel gears.

November 1, 1934, Portsmouth Navy Yard commenced the construction of the first two submarines *Plunger* and *Pollaik* employing Fairbanks, Morse engines of this type to drive generators for electric propulsion. Eventually the welded frames of the original design had to be replaced by an improved design and many engines of this type were built in 8, 9, and 10 cylinder units for propulsion of submarines and auxiliary surface craft. H.O.R. Company having acquired the MAN license for U. S. A. after World War I entered the field with a double-acting two-cycle MAN engine of the same cylinder size as was developed for the auxiliary generator engines of the cruiser *Leipzig* of the reborn German Navy completed in 1929. The principal parts of the original test engine, line 25, were imported from Germany to expedite the type-approval tests. The first set of four American-built nine cylinder engines of this type driving generators—Figure 20—were installed in the submarine *Pompano* construction of which by Mare Island Navy Yard began November 1, 1934. Although much trouble was experienced with these first engines at the beginning and the submarine eventually had to be re-engined

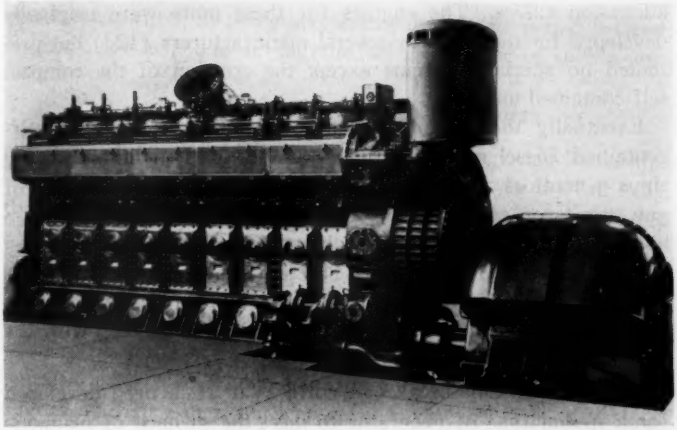

FIGURE 20.—HOR-MAN DOUBLE-ACTING TWO-CYCLE DIESEL ENGINE.

by improved engines of the same type, yet it operated with unusual distinction against the Japanese before it was lost—a high compliment to its engines as well as to its heroic personnel. A number of submarines and auxiliary surface craft were later propelled by these engines, and the incessant pressure of the war for greater speeds and therefore engine outputs and lower weight required several important modifications of design, such as a separately driven blower or twin blowers of higher speed. In the end however, this type of engine was declared as insufficiently uniform in performance to be used for submarine propulsion.

DIESEL GENERATORS FOR EMERGENCY, SHIPS SERVICE, MINE-SWEEPING.

Beginning with the 1933 cruiser and destroyer construction program, emergency generators were required in all combatant ships to supply emergency light and power circuits in case of breakdown or battle failure of the normal steam turbine driven ships service generators. Many different sizes were required from the original 25 KW sets for destroyers to 1000 KW for capital ships and all had to be automatically started by current failure actuating starting electric motors or solenoid-operated starting air

admission valves. The engines for these units were originally developed for the Navy by several manufacturers (12d) but presented no special problems except the creation of the compact self-contained units.

Eventually the advantages offered by the independent self-contained Diesel generator units were utilized in decentralizing ships generators and installing them in the immediate vicinity of gun turrets on capital ships, for instance. The same high-speed lightweight engines as used for submarine propulsion were driving these ships service generators. The remarkable thing is only the immediate success of their operation and the favor which they met on board the large combatant vessels as compared with the peace-time failure of former years.

Soon after the outbreak of World War II, a need developed for a special type of generator to meet the menace of magnetic mines by a new method of mine-sweeping. Hundreds of wooden hull twin-screw YMS mine-sweepers (18) propelled by two GM model 8-268 engines—line 26, Table I—carried a third engine of the same type which drove a special type of generator—Figure 21—to produce periodic magnetic impulses through hundreds of feet of electrically charged cable being dragged by the mine-sweeper.

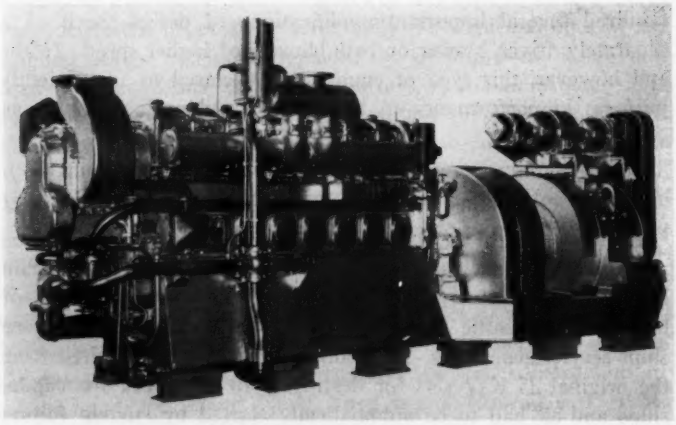

FIGURE 21.—GM-CDED MODEL 8-268 ENGINE DRIVING 540 KW MINE-SWEEPING GENERATOR.

A heavy flywheel, by being accelerated during the period between the impulses, helped to develop the 540 KW rating of such a unit.

Diesel Drives For Surface Warships.

The much discussed Diesel-propelled German pocket battleships were completed in 1933, 1934 and 1936 (12d). In the fall 1935, the new Chief of the Bureau—Rear-Adm. H. G. Bowen ordered a study to be made of a proposed 150,000 SHP. Diesel propelling plant for a battleship and the next spring the Congress was requested to appropriate $600,000 for the construction of experimental Diesel engines with the same cylinder dimensions as in the German battleships. Designs of three of the world's outstanding types of large size double-acting two-cycle Diesel engines were offered by their American licensees—American Locomotive Company had Sulzer Brothers license, HOR the MAN and Nordberg the Burmeister and Wain. In August 1936 each received a contract for the construction of a three-cylinder experimental engine. Eventually each engine passed the shop tests satisfactorily and qualified its maker to bid on the propelling engines of the U.S.S. *Fulton* a 9,734 ton, 18 knot submarine tender, building of which was assigned September 1, 1938 to the Mare Island Navy Yard. The American Locomotive Company was awarded the contract for the seven cylinder Sulzer-type engines of 4200 BHP. at 420 RPM., line 27, with each cylinder driving its own double acting scavenging pump from the crosshead. The engines have welded frames and the crankshaft is held in underslung cast steel girders under the bearings. Peculiarly enough the second engine gave a lot of trouble on the test stand through cracks in the frame which were eliminated only by removal of welded brackets for supporting lower cylinder heads when replacing them and by eliminating surface scratches and minute cracks around the raised coaming of the crankcase doors. The delay occasioned thereby resulted in the substitution of a Diesel-electric (D.C.) propelling plant for the Fulton (23) and the eventual installation of these engines in the oiler *Maumee* the hull of which was found in satisfactory condition to warrant extensive modifications including re-engining, so that the ship was able to serve reliably in both world wars.

The heavy burden of the contemporary development of the

modern high-pressure steam propulsion for surface Naval vessels and the imminence of outbreak of the World War II postponed and later completely eliminated any further use of the double-acting, two-cycle engine of large output.

In the meantime, the American Blower Company was apprised of the Bureau's needs for hydraulic clutches and acquired the manufacturing rights for same, and several reduction gear manufacturers were interested in the design of suitable gears (16). Later a considerable number of gears with hydraulic clutches were installed in submarine with a so-called "composite drive," having one engine geared to the propeller shaft through a hydraulic clutch with another driving a generator and transmitting its output to motors likewise geared to the propeller shaft. The first class of these submarines was contracted for September 19, 1935. Eventually geared submarine type engines were used in many auxiliary surface vessels, the first of which were the seaplane tenders contracted for September 1, 1938. Later on the alternating current Diesel-electric drive using the same submarine engines was developed for U.S.S. *Sperry* (contract signed June 12, 1940) another submarine tender and a sister ship of *Fulton*.

December 7, 1942 was not only the first anniversary of the Pearl Harbor Day, but also the launching day of the first two destroyer escort vessels, or DE's (1150 tons) at the Navy Yard Boston, propelled by General Motor's submarine Diesel engines through a D.C. electric drive. A few days before—November 29, 1942—the Consolidated Steel Corporation Ltd., Orange, Texas, launched its first DE propelled by Fairbanks, Morse submarine engines through gears. Hundreds of these vessels were built during the war.

The first LST (landing ship, tank) a storied giant of our landing craft program was launched at the Neville Island, Pennsylvania, plant of Dravo Corporation on September 7, 1942. It was a 327 foot twin screw vessel of 1490 tons (light) and was propelled by two model 12-567 engines—line 31, Table I—built by the Electro-Motive Division of General Motors Corporation each driving a reverse and reduction gear through an air-operated FAWICK friction clutch supplied by Falk Corporation. Over 500 of these vessels were built for the war.

Turbocharging Equipment Made Available In U.S.A.

The progress made with Buchi Turbocharging abroad was being watched with great interest by the Bureau and efforts were being made to induce American Diesel Manufacturers to acquire the patent and manufacturing rights. Eventually the American Locomotive Company built the first American turbocharged engine (6 cyl $12\frac{1}{2}'' \times 13''$) which completed its Navy type-approval test October 6, 1936 (15) using imported Brown-Boveri turbocharger.

This Company later organized its own production of that particular size of turbocharger. The first experimental turbocharged eight cylinder Cooper-Bessemer engine with an imported turbocharger was purchased for service test in a submarine. Winton Engine Company also converted and tested one of their eight cylinder engines. But the real impetus to this development was given by the Elliott Company's acquisition of the American manufacturing rights from the Buchi Syndicate in 1940 at the urgent insistence of the Bureau, and with its assistance in the conduct of the negotiations hampered by the war in Europe.

The period immediately preceding Pearl Harbor Day was used by the Bureau for developing new prototypes of ships and equipment which later were used in great numbers in the war. On June 15, 1939 for instance, a contract was signed for the construction of the first 170 foot submarine chaser—PC541, to be propelled by a new type of four-cycle turbocharged engine developed by Winton Engine Division of General Motors Corporation —Model 258S—Figure 22 and line 28 Table I. Two turbochargers were used—one each end and for each bank of eight cylinders. The engine was directly reversible and for sheer elegance of appearance and compactness of design it was difficult to match, but its eight hot exhaust manifolds presented such a large radiating surface that in a twin-screw vessel the operators would not linger in the passageway between the engines. Several different designs of asbestos-lined shields and enclosures were tried during the production period of these engines. The engine has a welded frame the upper part of which was used as a receiver for the charging air supplied by the blowers of the turbochargers. Eventually steel fabricated liners were used with porous chrome wear-

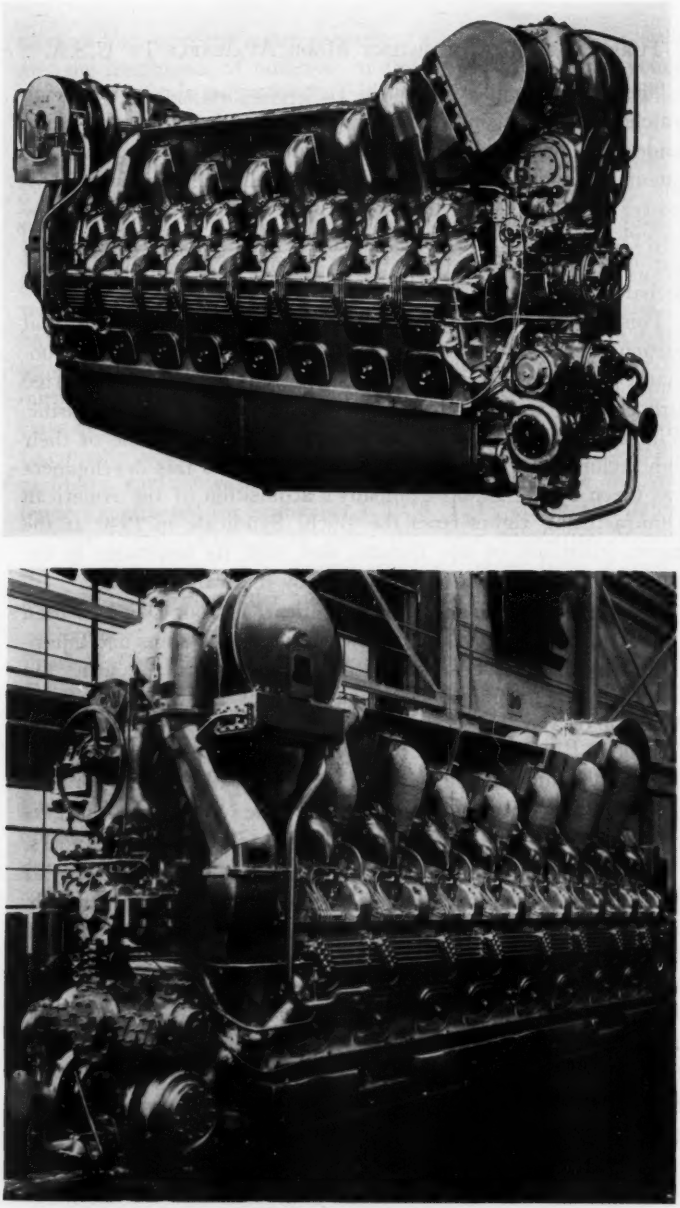

FIGURE 22.—GM-CDED MODEL 16-258S FOUR-CYCLE TURBOCHARGED DIESEL ENGINE FOR PC BOATS.

ing surface and several hundred of these engines were built during the war and made General Motors (CDED) the largest producer of turbocharged engines in the world to date.

The Bureau has also used a large number of the first turbocharged engine developed by American Locomotive Company out of their standard 12½ × 13 four-cycle engine used entensively in railroad locomotives. A pair of such engines of special design employing a light-weight welded steel frame were installed in one of our smaller submarines, used for training purposes during the war. Production of standard type Alco engines was also organized at the Busch-Sulzer plant and many of them were built there. The fuel injection pumps of this engine are immediately below the two hot exhaust manifolds leading to the turbocharger and on at least one occasion a ruptured fuel line caused a fire in the engine room besides burning many an awkward engineer in heavy weather. As a result, a water-cooled manifold was designed by the author and installed on many of these engines. This design is now commercially available under the trade name Helixhaust.

"Pancake" Engine Is Developed.

Perhaps the most outstanding design of Diesel engine developed for the Bureau was the "Pancake" engine. It was designed under the leadership and inspiration of "Boss Kettering" with Rear Adm. Bowen providing Bureau funds for its construction. The first test engine was built by the Research Division of General Motors and eventually completed its acceptance tests on October 31, 1940. Later the engine was produced in considerable numbers by the Electro-motive Division of the Corporation. It was the lightest Diesel engine used by the Bureau, it weighed only 4½ pounds per horsepower at its full-power rating of 1200 horsepower and it was the most radical departure from the prevailing practices. A vertical four-throw crankshaft with four pistons in each horizontal layer connected to each crank constituted its characteristic design feature and a 2 to 1 bevel reduction gear transmitted its output to the horizontal propeller shaft. The upper end of the crankshaft drove a centrifugal blower. These engines (18)—Figure 23 and line 29—were installed in the twin-screw 110 ft. wooden subchasers the first of which—SC453—made

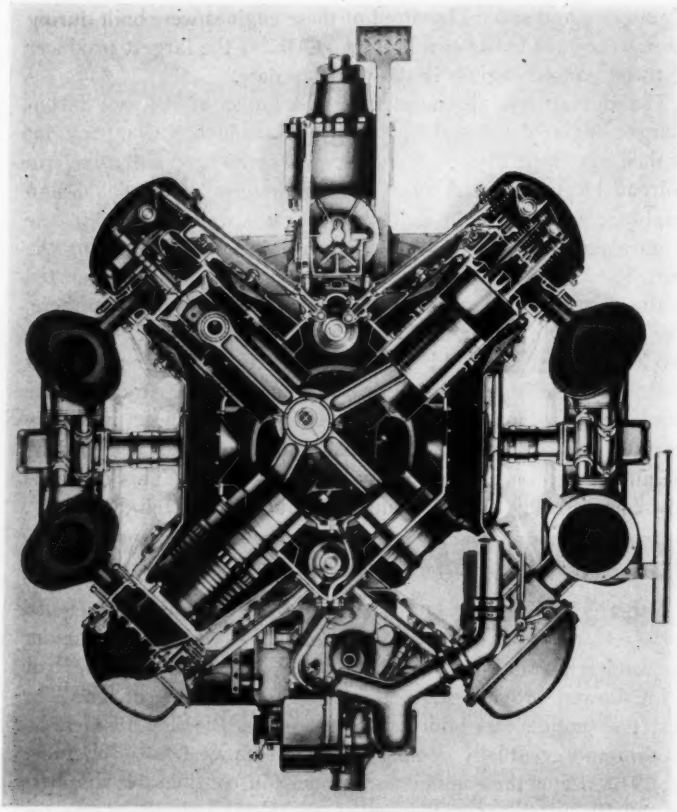

FIGURE 23.—GM PANCAKE (MODEL 16-184) TWO-CYCLE DIESEL ENGINE FOR 110 FT. SUBCHASERS.

a momentous 2300 mile trial trip from Detroit to the Brooklyn Navy Yard via the Great Lakes and St. Lawrence River in September 1942. Hundreds of these subchasers served in the war on convoy duty between Southern United States and South American ports as well in Alaskan waters near the Arctic circle.

The engine being non-reversible a bridge controlled variable pitch propeller was developed by GM Research Division and produced by Electromotive Division for all SC boats and later also used in the large infantry landing craft LCI(L), see (24).

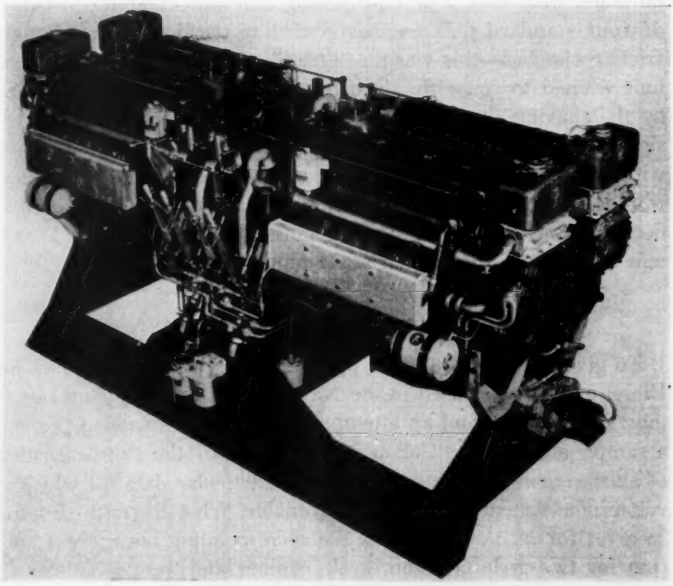

FIGURE 24.—GM QUAD ENGINE FOR LANDING CRAFT.

ONE TYPE ENGINE FOR ALL LANDING CRAFT.

The most distinguished service rendered by the Diesel engine in World War II was the ability of one and the same engine to power almost all of the Landing Craft despite their variety of size and purpose. This engine was the GM Model 6-71 built by the Detroit Diesel Engine Division, originally known as the Gray Marine engine because that company fitted it with a reverse gear and otherwise adapted it to marine service. One engine of this type was used in 36 foot long LCVP's and LCP's for vehicles and personnel over 56,000 of which were built during the war. Two engines were used in twin screw 50 foot and 56 foot LCM's for mechanized equipment, with a total of over 10,000 boats. Three engines were installed in triple screw LCT's for tanks, 114 and 119 ft. vessels of 285 tons loaded, of which about 1500 vessels were built. To propel the LCI's for infantry—a 159 ft. twin-screw vessel of 387 tons loaded displacement—of which over one

thousand were built a new propelling unit was created consisting of four standard 6-71 engines geared to one bull gear through friction clutches—this was the "Quad". The gears of the original unit whined to rouse the dead but later better gears were made by the shaving method. The clutches were installed by the designer probably in the secret hope that they would not be used with the unit running, and all were agreeably surprised when they were tried on the test stand under full load to connect and disconnect any one of the engines. A most flexible propelling Diesel unit was thus created which never failed the ship in need.

Diesel Engines In Wartime Navy.

To give an adequate word picture of the inevitable variety of Diesel engine types used in the Navy during the war would be a most difficult task and an attempt was therefore made to present a sample analysis in "Gallop Poll" fashion of the characteristics of all the engines in the service on one particular date for which a voluminous statistical report was available. This diagram—Figure 25 gives for each cylinder bore the corresponding horsepower and rpm for two-cycle and four-cycle engines and the percentage of total horsepower represented by each such bore.

Over fifty million horsepower of Diesel engines had been built during the war for the Navy. Capt. L. F. Small of the Bureau of Ships in his speech before the SAE Diesel Meeting Chicago (19b) announced "that as of October 1943 the combined horsepower of Diesel engines being installed in Naval craft exceeded that of the steam turbines for the first time in the history of our Navy."

The research devoted to the development of more suitable designs and types of Diesel engines (12e) which was sponsored by the Bureau was tremendous but not all of it, quite naturally ripened to the stage of affecting the war production of engines. It is obvious of course that the Bureau was not designing any of the Diesel engines built for U. S. Navy but by keeping abreast with their development everywhere it attempted to foresee the direction in which competent designers would be willing to undertake the responsibility of the next step to improve the suitability of the Diesel engine to solve an increasing range of service require-

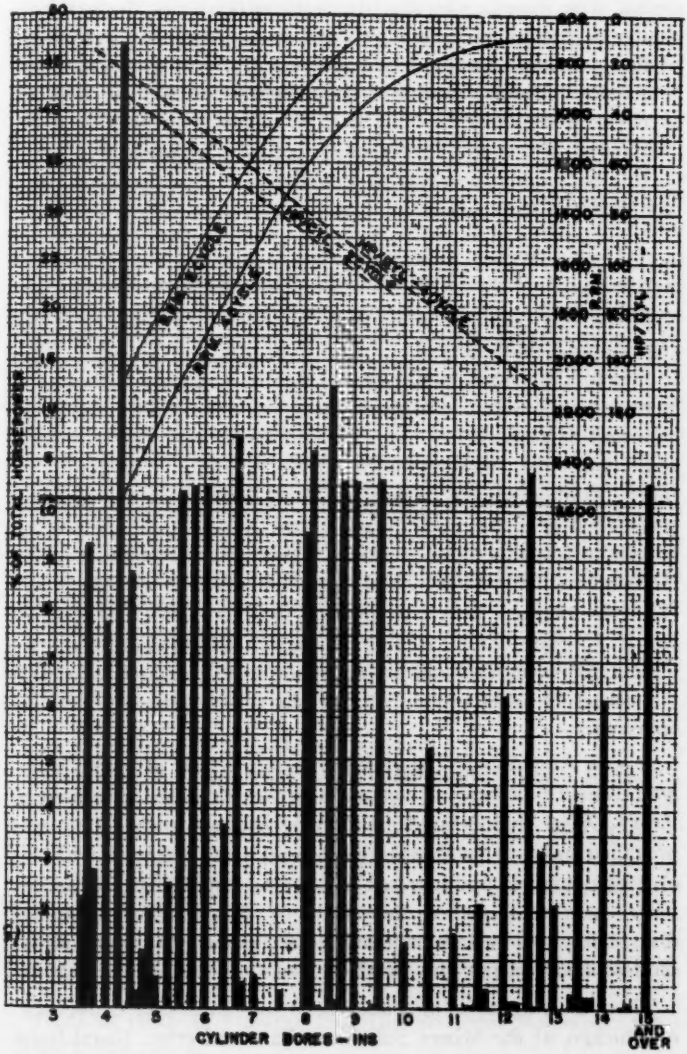

FIGURE 25.—BHP, RPM AND PERCENTAGE OF TOTAL HORSEPOWER FOR DIFFERENT CYLINDER BORES OF DIESEL ENGINES IN U. S. NAVY.

ments. The Bureau was also the rallying point for the patriotic engineering talent of the nation eager to cooperate in the solution of the problems of our Navy under the stress of grave national emergency. And as example of such cooperation may be cited the excellent work done by the SAE War Engineering Board on "Improved Means for Evaluating Effects of Torsional Vibration on Internal Combustion Engine Installations" which resulted from one such call for cooperation (19a).

Men Not Engines Win Wars.

In the final analysis it must be conceded that men and not engines and machinery are capable of the will to win. This tribute to the Diesel engine would not be complete without the mention of the heroic effort which was made to train thousands of men and develop within them the confidence required to intelligently meet the emergencies of engine operation. Thousands of college graduates were subject to Diesel "indoctrination courses" established at Penn State, Cornell and N. C. State engineering schools to form the core of Naval reserve officers entrusted with the operation of the vast Diesel propelled Navy. Eventually many full-sized engines were installed in the laboratories at Cornell and N. C. State (25) involving millions of dollars in order to give these embryo engineer officers an opportunity to match wits with the actual "trouble gremlins" that thrive on ignorance and inexperience.

A vast Amphibious Diesel Engineering School at Flint, Michigan trained thousands of operators for the landing craft engines. Every Diesel engine builder supplying engines to the Navy was prevailed upon to establish training quarters equipped with his own type engines and manned by his own engineering personnel, where men were getting the "feel" of the parts by practicing disassembly and replacement of selected parts and stood watches on test floor during the gruelling tests of production engines.

There are thousands of graduates of the "Steel Pier University" of Chicago, of the Miami Subchaser Base and other Naval training schools which now run the Diesel locomotives, trucks and tractors of our nation in peace.

The story of maintenance and of providing spare parts for the Diesel engines scattered as they were all over the world is an epic all its own covered by Captain C. S. Seabring elsewhere and by Commander T. G. Reamy earlier in the war (24).

The Chief's "Well Done" To Diesel Engine.

Vice-Admiral E. L. Cochrane, the wartime Chief of The Bureau of Ships and an able shipbuilder and administrator, speaking before the Navy League at Milwaukee, Wisconsin on Navy Day in 1945 stated that at the end of the war there were "1312 combatant ships on hand or under construction" and "10 million tons of auxiliary ships and landing and other supporting craft on hand". On September 12, 1947 speaking before the Regional Conference of Diesel Engine Manufacturers Association in Boston, Massachusetts (21) he concluded with the following tribute to the Diesel engine. "The United States, in this last war, built thousands of ships and 90 per cent of them were Diesel powered. There was no vessel that was not equipped with Diesel power of some kind; much use was made of Diesel auxiliaries. The whole landing craft program was Diesel powered, and these ships worked under the most strenous conditions".

"In many an emergency, when a ship was bombed, Diesel engines saved the day. Not a single one failed us in a critical operation."

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SPARE PARTS—ENIGMA OF WORLD WAR II.

By Captain C. S. Seabring, U.S.N.

The Author.

Captain Seabring was graduated from the Naval Academy in 1927 and served in cruisers, destroyers, and submarines. An engineering PG, he was designated EDO in 1941. From 1942 through 1944 he was Head of the Internal Combustion Engine Section of the Bureau of Ships. In 1945 he was Executive Officer of the Naval Technical Mission in Europe, and presently is Head of the Department of Marine Engineering at the U. S. Naval Academy.

One of the most controversial subjects in the Diesel field during World War II was the scheduling, production and delivery of spare parts. No one "in the know" will deny that many more parts were required than could actually be put in service, but, conversely, no one "in the know" will ever feel our sights were set too high. Filling the supply pipe lines necessary to maintain our materiel in required state of readiness in order to effectively wage a global war was, in itself, a gargantuan program, but not until its accomplishment was more than a trickle of spares available at the operational ends of the network.

Because of the very nature of spare parts, the subject is apt to be boring, and to some this article will prove no exception. The great mass of detail required for intelligent administration of a coordinated program is so overwhelming that many people are inclined to dismiss the subject with a wave of the hand.

The magnitude of the maintenance problem can probably be better understood if we stop to consider that at the beginning of 1942 there were only 600,000 Diesel and gasoline horsepower in service. 9,030,000 horsepower were added during that year, and in 1943 an additional 19,937,000 horsepower were actually delivered. The maintenance load at the beginning of 1944 was therefore almost 30,000,000 horsepower. Predicted new construction for 1944 was 21,906,000 horsepower and for 1945 22,437,000 horsepower. The predicted maintenance load at the

outset of 1945 was consequently over 51,000,000 horsepower. Expressed in another way, the steam maintenance requirements for 1944 were increased by 63% over 1943 requirements, while the Diesel requirements increased 226% and the gasoline requirements 185% for the same period.

The first Diesel engines were introduced into Naval Service sometime prior to the First World War. From then until the declaration by the President in 1941 of the Naional Emergency, most of the Navy Diesel engines were installed in submarines. Such engines were generally custom-built to Navy specification and procured in small quantities complete with on-board, tender and overhaul spares. The overhaul parts were marked for each ship and stored in lockers at their home bases.*

Vessels were scheduled for definite periodic overhauls at specific yards and the replacement parts requirements could be anticipated well in advance. When such needs developed parts were requisitioned through Naval Procurement Officers or purchased directly from manufacturers by Supply Officers of the ships or overhaul yards concerned.

Spare parts were identified by plan and piece number on the contract under which the engines were originally procured. Identification was limited to specific engines and did not apply to engines purchased under other contracts. Most of these manufacturers had not developed part number systems even so far as to cover identical parts on different engine models.

Inasmuch as raw materials were plentiful and the Diesel Industry was just coming into its own and as yet not overburdened with large construction programs, small maintenance parts orders placed by the Navy could be quickly produced.

Engineering Officers and men were well trained in the proper operation and maintenance of Diesel engines, and peacetime maneuvers normally did not subject the engines to long hours of operation under heavy loads at high speeds.

It can be readily seen that under such circumstances no great difficulty was experienced in supplying Internal Combustion maintenance parts nor was controlled global distribution essential.

Immediately after the outbreak of War, the Diesel Industry

*The information on spare parts incorporated in this article is a resume of the official records of the Ships Parts Control Center, Mechanicsburg, Pa.

was called upon to produce complete engines on a schedule that taxed manufacturers' production facilities to the maximum.

Many new engines were introduced into service in order to meet the demands of the accelerated shipbuilding program for propulsion units in new type surface craft. A similar situation existed with manufacturers of marine gasoline engines. So urgent was the need for complete units that the production of new engines was scheduled to the full capacity of the Industry's output with little regard to on-board or maintenance parts requirements.

Only because new vessels could not be commissioned without complete allowances of spares, were engine production schedules finally relaxed enough in the early part of 1943, to allow manufacturers to produce just enough spare parts for on-board allowances. Only in the cases of extreme emergency breakdowns that had to be fully substantiated, were maintenance parts given overriding priority.

Even with Internal Combustion engine manufacturers' plants completely turned to the output of new units the immediate demand for vessels to combat the submarine threat in the early days of the War could not be entirely met. To supplement the construction program the Navy took over and converted numerous pleasure and commercial craft for patrol or convoy service. Thus, the Navy inherited a collection of miscellaneous engines of various makes and models. Many of them had long since been out of production and needed parts to put them in such condition that they could withstand the rigorous assignments demanded of ships in Wartime Naval Service. More often than not neither bills of material nor drawings were available with which to identify or determine replacement parts requirements. Some sources of supply had completely dropped out of the picture and could only be re-established after long weeks of engineering research.

Along with the inflated requirements for new vessels came the equally important need for personnel to man them. This necessarily limited training given new crews and their unfamiliarity with proper engine maintenance had a telling effect on replacement parts demand.

Financial limitations for ships maintenance, previously controlled by quarterly allotment, were withdrawn and there imme-

diately followed a period of uncontrolled and indiscrimate spare parts purchases. Manufacturers were swamped with small orders from ships and repair bases, some requesting exorbitant quantities in an attempt to hoard items that were known to be critical and frequently requiring replacement. A large percentage of these orders could not be interpreted due to insufficient information and unintelligent requisitioning practices. Such was the spare parts picture in March of 1942.

It became apparent that this method of spare parts procurement and distribution was entirely inadequate to meet the needs of a vastly inflated fleet so widely dispersed. Controls and the establishment of an orderly program were needed at once, unless ships maintenance was to bog down completely. It was essentially for these reasons that the Bureau of Ships and the Bureau of Supplies and Accounts by joint letter on 13 March 1942 established the U. S. Navy Diesel Supply Depot, Mechanicsburg, Pennsylvania, as a clearing house for all engine parts requsts. On 25 July 1942, the name was changed to Diesel Department, Naval Supply Depot, Mechanicsburg, Pennsylvania, and the responsibility expanded to take in the handling of requisitions for gasoline as well as Diesel parts.

Although no stock was on hand by 3 September of that year, Mechanicsburg, at least as far as paper routine, was in a position to start operations under the new plan. All manufacturers of Internal Combustion engine parts were immediately instructed to no longer accept direct orders or negotiations for spare parts. Fewer orders for larger quantities that had been interpreted as well as possible with the technical information available could now be placed with manufacturers in place of thousands of small miscellaneous requests. Although this was a tremendous step forward, the real job had just begun, and many complex problems had to be overcome before any semblance of order could come from the confused situation.

While Mechanicsburg had been set-up to receive and screen all orders for engine parts, consideration had to be given to anticipating future maintenance requirements so that adequate stock piles would be on hand as needs developed. Tabulated records of numbers of each engine model in use and estimated annual hours of

operation were used as the first guide in placing overall stock contracts. Sums of money based upon estimated horsepower of engines known to be in service were set aside for the procurement of spare parts for each manufacturer's model. Manufacturers were asked to submit recommendations of items and quantities to be procured up to the dollar value allocated for their engines. Emphasis was stressed on those parts known by manufacturers to require frequent replacement under similar operating conditions. Quantities ordered under these contracts were intended to serve only until a more reliable procurement policy could be developed. Negotations were completed in the first months of 1943.

By this time engine construction was in high gear and still had preference over the production of spare parts. The scheduled lead time for many badly needed parts ordered under the new contracts was as high as six and eight months. It became evident that only comparatively small quantities would be available for distribution in the immediate future.

At the same time requisitions for engine parts were being received by Mechanicsburg in increasingly greater volume. New ships were coming into service daily and old ones needed attention to keep them operating. In addition, as operations expanded and the War took on a more global aspect, the need for shortening supply lines to meet emergency conditions and maintain full striking power of the fleet at all times was vital. To meet these various necessities the Automatic Flow Plan for global distribution was designed.

The Automatic Flow Plan was undertaken with the following primary objectives:

(1) To establish a sound procurement policy based initially on approved procurement factors and later by consideration of actual global usage experience as the information became available.

(2) To shorten global supply lines by the establishment of strategically located bases where small piles of spare parts could be maintained thereby reducing long shipping time on emergency demands for advanced areas.

(3) To assemble, catalogue, and distribute parts books, parts lists, drawings, and such other technical data as might be necessary for the proper interpreting and ordering of engine parts by ships or field activities.

(4) To ration the flow of available material on certain high production engines by the distribution of monthly spare parts allotments to Spare Parts Distribution Centers, based on the number and types of vessels operating within that area. These allotments to be anticipated in sufficient quantity to meet day to day requests and allow for gradually building up small stock piles of parts most frequently requisitioned.

The first step in determining the most intelligent approach to the problem was a complete analysis of the tabulated lists prepared from questionnaires sent to all vessels of the fleet. As a result of this study these findings and decisions were made:

(1) 556 different models of engines were at that time either in service or scheduled to go into service.

(2) Nearly one-and-a-half million different identifying markings of plan and piece number, manufacturer's part, or production number and accessory manufacturer's number were found to be in use.

(3) 128 of the 556 different engine models represented 93% of the total horsepower.

(4) A breakdown of the parts in the 128 models into "FAST Moving", or frequently replaceable parts, and "SLOW Moving", or parts not often replaced, indicated that 15% of the items of the Fast Moving Group would account for about 80% of the total urgent maintenance requirements.

(5) Having thus determined that by concentrating on some 15,000 items the bulk of the job could be accomplished, these were the parts selected for Automatic distribution and the ones on which close scheduling and reporting procedures would be set up. Because of the high usage and large quantities required, they generally speaking were the most critical items.

Parts lists were put into the hands of the manufacturers of the 128 engine models who were asked to recommend a procurement

factor for each item on each model engine. This factor was to represent his best estimate of the number of pieces required to keep one engine operating for one year. No reliable information was available as to the number of anticipated hours of operations or the conditions to which engines would be subjected. Only by carefully studying performance data for each part under what was thought might be similar operation conditions could any where nearly accurate figures be arrived at.

In addition to procurement factors, the manufacturers were to work up numerical interchangeability lists to show the different models that each part would be used on, and the quantity installed in each type engine.

The Master Control Center which had been set up at Naval Supply Depot, Mechanicsburg, prepared decks of tabulating cards for each engine model that indicated the following information as furnished by the Bureau of Ships:

(1) Part Number
(2) Nomenclature
(3) Manufacturer
(4) Model Application
(5) Number used per engine
(6) Procurement factor
(7) Fast or Slow Moving Part

By taking projected figures of engines expected to be in service by the end of the year 1943 and multiplying them by the procurement factors in engine model cards, a one year's requirement for each part was arrived at.

It was then necessary to compare these quantities with those already under contract so that deficiency contracts for the items that were under-bought could be immediately negotiated.

So that the Master Control Center could act as a central administrative agency for all engine parts contracts, unshipped balances on Bureau orders previously placed for base spares, Lend Lease material, and repair base stock requirements were placed at the disposal of Naval Supply Depot, Mechanicsburg. Shipping instructions were modified to bring this material into Naval Supply Depot, Mechanicsburg, or such other destination as Mechanicsburg might designate. By this action both procurement and dis-

tribution problems were considerably complicated in that Mechanicsburg was not properly notified of undelivered balances in many instances prior to the receipt of material.

It was agreed that eight strategically located major supply points or Spare Parts Distribution Centers would be set up to receive, store, distribute and account for engine spare parts in their areas. The distribution to SPDC's of fast moving parts was to be made by monthly allotments based on the number of engines of each type operating within a given area, procurement or usage factor and a 1.75 factor to compensate for shipping losses and the building of small stock piles.

Information received from overseas activities indicated that a large percentage of parts being received were not in usable condition due to improper preservation and lack of adequate identification. In making shipment to Spare Parts Distribution Centers in advance areas consideration had to be given to box sizes and weights.

Box specifications would have to be such as to make them serviceable as temporary bins in many locations. Further, box weights should not exceed a maximum of two hundred pounds as many of them would have to be moved from beaches to storage points by hand. The standardization of box markings was most important in order to expedite the location and movement of material through trans-shipment points.

The Bureau of Ships in cooperation with the packing and shipping section at Mechanicsburg set up standards for preservation, packaging, boxing and marking of material under Navy spare parts contracts. All manufacturers participating in the Automatic Flow Program were personally contacted to assist them in setting up to meet the new standards.

One of the most important programs undertaken in connection with Automatic Flow, was the training of SPDC personnel. Before SPDC's could start functioning complements of officers, trained parts men, and technicians, had to be indoctrinated into operations such as stock control, storage, proper requisitioning practices and the use of engine parts books, as well as other interpreting media.

In July, 1943, eight Spare Parts Distribution Centers were established in the various geographical areas as follows:

SPDC CODE	*LOCATION*
Left	Brisbane, Australia
Epic	Noumoa, New Caledonia
Fray	Pearl Harbor, Hawaii
Glen	Oran, Algeria
Pith	Trinidad, British West Indies
Fear	Recife, Brazil
Join	Dutch Harbor, Alaska
Jowl	Exeter, Devonshire, England

In addition, Naval Supply Depot, Mechanicsburg was designated to exercise the functions of a SPDC for continental bases including Panama, but excluding Hawaii and Alaska.

The British having acquired numerous Diesel propelled craft through Lend Lease appropriations had to be considered in the distribution of maintenance parts. Rather than attempt to compute quantities and deliver material direct to the various bases under Automatic Flow in England, total British requirements were shipped to one central distributing point from which the material was fanned out. Two other Central distributing points were established in 1944 at Malta and Bombay, India. Automatic Flow distribution was extended to also take in the Lend Lease governments of Canada, Russia and Brazil. Smaller requirements for numerous other Lend Lease vessels were supplied through the various SPDC's. Engine population figures provided each month by the Bureau of Ships indicated the number of foreign government units in operation in each SPDC area. Monthly allotments were increased to allow for maintenance of these engine models.

Although all of the planning had not been completed, the first distribution of engine parts under the Automatic Flow Plan was made in June, 1943, on the following General Motors, Cleveland Diesel Engine Division models:

(1) 3-268A
(2) 8-268A
(3) 8-268

Other models were introduced into the program in subsequent

months with the plan scheduled to be in full operation by the end of 1943.

Under this system of world wide distribution involving large quantities of parts for so many different engine models, the burden of making monthly overseas shipments would necessarily have to fall on the manufacturers. Mechanicsburg packing and shipping facilities were known to be entirely inadequate for a program of this magnitude and the flow of material might be dangerously bottle-necked if it were attempted. Every month, therefore, the Master Control Center computed specific shipping quantities for each SPDC and submitted them in tabulated form to various manufacturers for direct shipment of the material from the factory. Due to the lead time and the negotiation complications involved in placing new orders a system of diversions from large stock contracts placed in anticipation of future requirements was worked out in the handling of monthly allotments. A letter accompanying the shipping instructions served as an authority to divert for direct overseas or continental shipment quantities as indicated from any open stock contract.

As a precaution against the building up of excess stocks at any one point, definite stocking levels based on geographical locations were established for each SPDC by the Bureau of Ships in April, 1944. Once a base had attained this level on any given part, automatic shipments were temporarily discontinued regardless of issues until the balance had dropped below the allowable stocking limit.

To compensate for inaccuracies, quantities distributed on a procurement factor basis due to unpredictable operations and other unusual circumstances, SPDC's could request on the monthly report to Master Control Centers quantities required in excess of Automatic Flow allotments, providing the specific need was substantiated.

As usage data was received from the field it became more and more apparent that no distribution formula could be worked out to accurately anticipate requirements where only a small number of engines were in service. Fast Moving parts for a number of models were, therefore, removed from automatic monthly distri-

bution and handled by quarterly allocations based on usage and balances on hand.

One of the most troublesome problems in the distribution of engine parts was the continual superseding of part numbers by manufacturers due to design and substitution of material. Some changes in design made as a result of excessive operation failures were necessary and entirely justified. Many others, however, were made indiscriminately. Unestimatable thousands of man hours were expended in attempting to keep SPDC's informed on part number supercessions, thereby assisting them in requisitioning and stock problems of identification.

Many items could not be scheduled in nearly sufficient quantities to meet monthly shipping requirements. Generally speaking, these parts were also badly needed for new engine construction which still in 1944 had priority over the production of maintenance spares.

As a result, monthly shipments were going forth to SPDC's minus such critically needed material as shell bearings, ball bearings and piston rings. The Bureau of Ships in July, 1944, directed manufacturers to give current months allotments preference over previous months shortages. Inasmuch as some parts were never at best produced or made available to the Navy in quantities great enough to more than meet current months demands, many shortages created early in the program were never satisfied. Not until a War Production Board Directive issued in November, 1944, giving replacement parts priority over new engine production, were critical materials made available in sufficient quantity to meet Automatic Flow monthly allotments.

As the war progressed in the Pacific, fleet operations were greatly accelerated and expanded. Some of the Spare Parts Distribution Centers originally set up in 1943 in strategic areas were in the space of a few short months several thousand miles away from advance units. This meant that new bases had to be established on short notice. Such parts were available at old locations and had to be moved up to the new bases. This material, of course, was frozen during the transfer period which in some cases was a matter of six to eight weeks. Mechanicsburg, upon notification by the Bureau of Ships that a new SPDC was to be activated,

computed stocking level quantities for all parts required to serve Internal Combustion engines assigned in the area. Items available at Mechanicsburg and Clearfield were shipped from stock to a pre-designated assembly point. Special shipping instructions for direct shipment to the assembly point were issued to manufacturers concerned for the balance.

Studies made of distribution time factors indicated that the elapsed time from the date an SPDC prepared and forwarded usage data to Mechanicsburg until shipment computed from these figures arrived at its designation was between seven and eight months depending upon the location of the base. It could not be expected under the circumstances that by the time monthly allotments were received they would correspond in quantity to prevailing usage requirements. Further, it had been firmly proven that SPDC's by virtue of their proximity to fleet operations were in a better position to anticipate their own future requirements than the Master Control Center at Mechanicsburg. In consideration of these facts monthly Automatic Flow distribution by formula was discontinued in August, 1945, in favor of a bi-monthly requisition plan, whereby SPDC's could estimate their own future requirements for a two months period.

Many individuals disagreed with the Automatic Flow Plan both in principal and operating detail. Even proponents of the system freely admitted that it had certain inherent weaknesses. Whether any other method of distribution could have been designed to do a better job under circumstances prevailing at the time is debatable and largely a matter of conjecture.

As a result of a very exhaustive survey of the Mechanicsburg operations made by Naval Personnel, principally of the Bureau of Ships and augmented by persons prominent in industry, it was determined that an Inventory Control Center should be created out of the cloth from which Mechanicsburg originally had been cut, to take over from the Naval Supply Depot the inventory control functions for material in the ships parts supply system.

Based on this survey, in June of 1945 the Ships Parts Control Center was created at Mechanicsburg, and designated as the supply-demand control point for Bureau of Ships Internal Combustion Engine and hull and machinery material. Early in 1946

the first draft of a plan for control of inventory of Bureau of Ships' material was presented to the Secretary of the Navy for his approval. This plan later was approved, and now constitutes the basic document for the operation of the Ships Parts Supply System. There were two major tasks to be performed under that plan by the supply-demand control point:

First, to effect complete identification of material in the Ships Parts Supply System, and

Second, through the medium of a stock status report to learn what material is in the system, how much there is and where it is.

The problem of cleaning up identification, therefore, was one of the most vital tasks confronting the SPCC at the beginning of its operations. Without proper identification, it could not be guaranteed that items supplied were, in fact, the items required. Without identification, much material that was in store in the various stocking points was not recognized as identical to material requested. Without identification, needless procurement was made of items to fill supposed deficiencies, when, in fact, adequate stocks under different names or numbers already existed. Without identification, much manpower and effort was expended in interpreting requisitions finally to identify the item that was required, so that issues could be made from stock or procurement made from manufacturing sources, as appropriate.

The second major task under the approved Ships Parts Inventory Control Plan is the responsibility for periodic stock status reporting to a central location.

In order for proper inventory control to be exercised, it is necessary that the principal stocking activities make periodic reports of stock status to the central control point. Once each quarter all of the activities directly supplied or dependent activities submit reports of stock on hand stock due in, and issues made over the preceding quarter. These reports finally are brought together in the Ships Parts Control Center in Mechanicsburg and consolidated into a single report for the entire system, showing the material on hand, material issued from the system, and providing the data on which estimates are made of the material necessary to bring the system stock up to authorized levels.

Potentially, the Ships Parts Supply System could be expected to supply in excess of two million items to meet maintenance and repair demands of the fleet. Actually, most of these are parts for which there has been little or no prior usage, and for that reason the items have not been regularly stocked in the Ships Parts Supply System. There are on hand in the Ships Parts Supply System two hundred twenty-six thousand different items which constitute those repair parts for which, over a period of time, there has been repeated demand. For the other one million seven hundred fifty thousand potential demand items, there has been some demand for limited additional items, but generally not a sustained demand to justify establishment of stocks of the items.

In order to stock an additional one million seven hundred thousand items for which infrequent demand might develop, it would take in excess of seven billion dollars of the taxpayers' money. Money of this character is not available. Even if it were, good, sound business practices would preclude its use for maintaining stocks of items for which little or no demand had previously existed.

THE OUTLOOK FOR NUCLEAR POWER.

By Cecil B. Ellis.

Introduction.

It is generally agreed that nuclear power could be applied to the generation of electrical energy and to the propulsion of large vechicles. However, it is now three years since the end of World War II and so far as is known no light bulb has yet been lit or engine turned by fission power. This paper will examine some of the grave engineering and economic difficulties in the way of nuclear power development—as far as seems possible without involving classified information.

The Author.

Dr. Cecil B. Ellis' record is: Ph.D. in vacuum spectroscopy from the University of Michigan in 1935; instructor in physics at CCNY and research in spectra of solids at Columbia until 1941; a year at the GE Research Lab. developing one-centimeter radar magnetrons; and the rest of the war on the Manhattan Project. He helped build the K-25 isotope separation plant at Oak Ridge—devising and helping to disseminate the new techniques of high vacuum and of large scale gas analysis with the mass spectrometer. Since the war he has helped to edit the 70-volume Summary Technical Report of OSRD and carried on nuclear research at Columbia University. He joined the Office of Naval Research in 1947, where he is now concerned with nuclear power and other nucleonics application in the Navy.

I. Description Of Fission Piles.

There are at least eight piles operating in this country at present, at Hanford, Los Alamos, Argonne, and Oak Ridge. These piles do not produce usable power, but were designed to make plutonium, to produce radioactive isotopes, or for research. Other research piles are in operation in Canada and England. Several new piles are now under design and construction both here and abroad, and some of these are expected to produce usable power on a demonstration scale.

In one simple type of pile air-tight aluminum cans, containing uranium metal, are imbedded at intervals throughout a large mass of pure graphite. See Figure 1. Cooling water is passed through

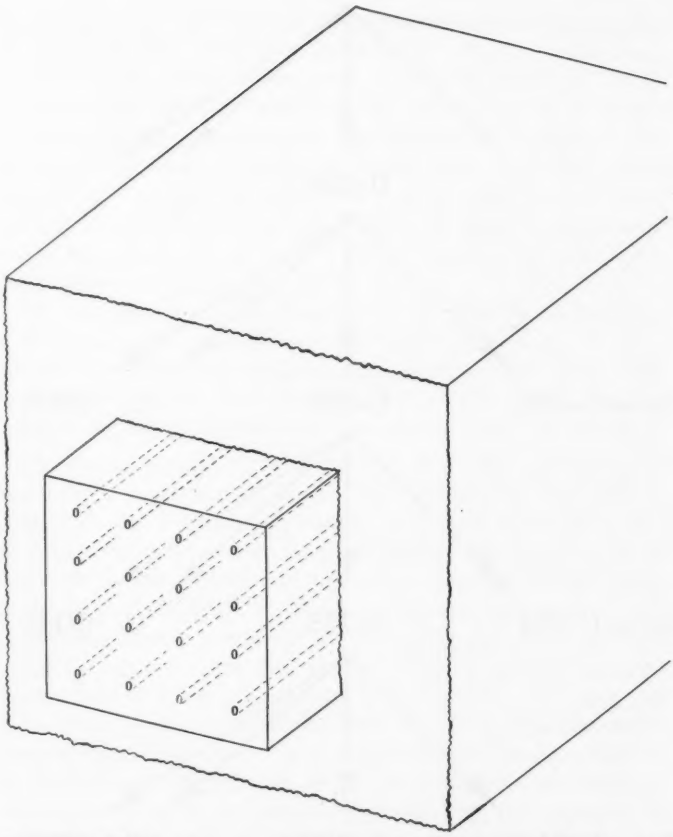

FIGURE 1.—Highly schematic view of one possible type of pile, showing concrete shielding cut away in front to expose the graphite pile proper, with cylindrical channels for the insertion of cans of uranium. A cooling fluid might also be passed through these channels.

channels around the uranium-filled cans. The entire assembly is surrounded by a concrete shield several feet thick to prevent the escape of damaging radiation.

The uranium metal contains a rare isotope, U-235, and a plentiful one, U-238. The normal ratio, as found in the ores, is 1:140. Slow neutrons will cause fission of the rare isotope, U-235, the

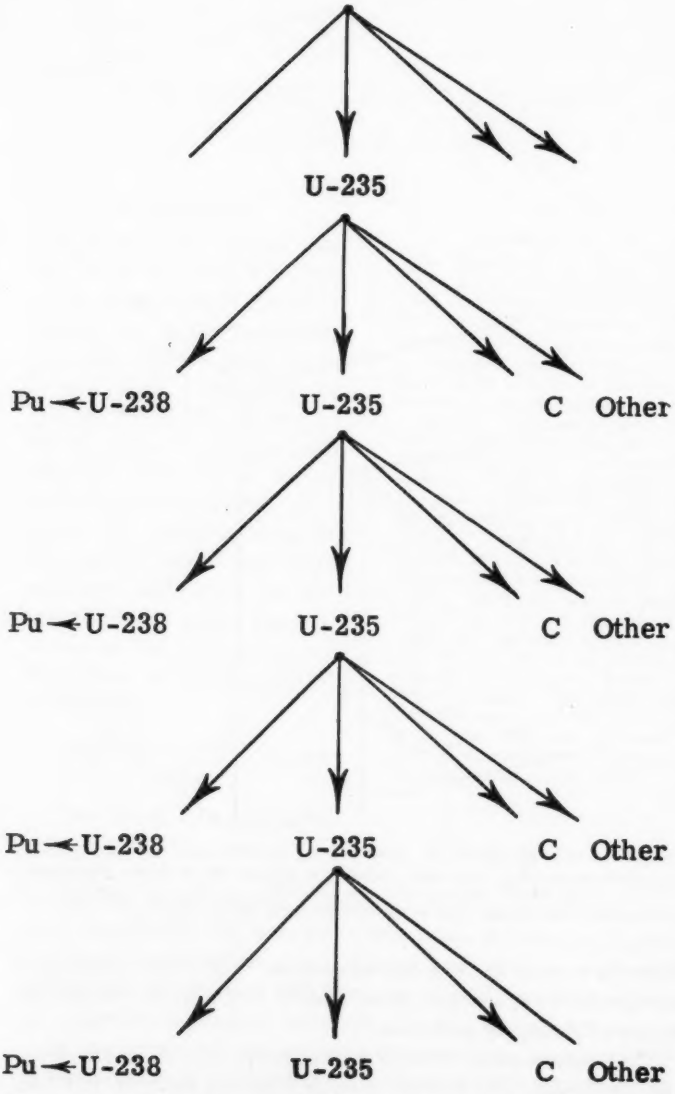

Fig. 2. Neutron behavior in the U-235 chain reaction.

nucleus of which splits into two heavy fragments flying apart with an energy of about 200 Mev. In addition to the two main fragments a number of neutrons result from each explosion. These neutrons come out of the fissioning nucleus with energies averaging about 2 Mev, according to the unclassified literature. The purpose of the graphite (called the "moderator") is to slow down the neutrons, by collision with carbon nuclei, to the energy range around 1/40 of an electron-volt. They will then be more easily able to cause fission in the next U-235 nucleus they approach, with the possibility of setting up a chain reaction.

The number of free neutrons resulting from the fission of a U-235 nucleus is probably not the same every time. It may possibly vary from none to half a dozen or more. But since many millions of fissions per second occur in the usual pile, the thing of importance is the *average number of neutrons per fission.*

The future history of these neutrons is of the highest importance for the operation of the pile. The situation is illustrated by Figure 2. The four arrows coming from each fissioning U-235 nucleus represent the four kinds of things which can happen to the emitted neutrons. Some of them will be captured by the carbon nuclei (C) of the graphite. Some will be captured by the U-238 nuclei, not causing fission but leading to the conversion of these nuclei to the new element plutonium (Pu). Some will be sufficiently slowed down by collisions in the graphite that they can cause fission in the next U-235 nucleus they strike. The remainder, lumped together as "other" in the diagram, may be (*a*) captured by impurities in the pile (any slight traces of boron or cadmium, for example, are strong capturing agents); or (*b*) captured by hydrogen in the cooling water, or by various elements in any structural members or instruments included in the pile; or (*c*) lost by escape through the outer walls into the shield.

The maintenance of the chain reaction depends on reducing the losses by capture or escape until, on the average, at least 1.0 neutron from each fission is available for the next fission. Usually what is done in practice is to reduce the capture losses as much as possible by proper choice of geometry and materials. And then to keep increasing the pile dimensions until the escape loss drops and the number of neutrons per fission left to make the next

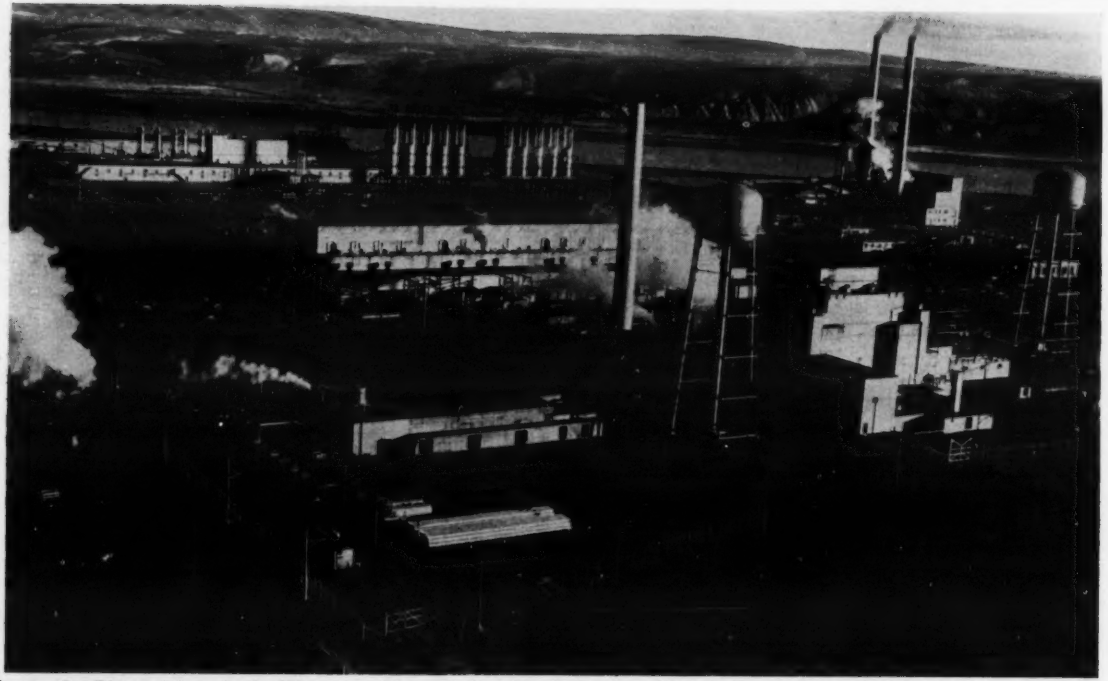

FIGURE 3.—Pictured here is the Plutonium Plant of the Atomic Energy Commission at Hanford, Washington. The plant has changed in the three years since this picture was taken, but the AEC has not released any more recent pictures which would reveal changes.

generation rises to perhaps 1.001. The pile has then passed the "critical size" and the chain reaction will rapidly grow. The system may now be stabilized at a desired power level by inserting into the pile a "control rod" containing a neutron-capturer like cadmium or boron. The multiplication factor may be reduced by this means from 1.001 to 1.000 and the number of fissions per second held constant.

The large piles at Hanford were built to produce plutonium, (Figure 3). The uranium cans are periodically removed from the pile and the plutonium, created from U-238 by neutron capture, is separated by extensive chemical processing. As already discussed, the U-238 capture rate must be kept rather small so as not to choke off the chain reaction. The result is that far less plutonium is generated per second than the amount of U-235 used to maintain the reaction.

So far in this description the attention has been on the neutrons while the two principal heavy fragments coming from the fissioning U-235 have been neglected. The velocity of these heavy fission fragments is so great, (total kinetic energy about 200 Mev), that their collisions with the other atoms of the pile result in a large amount of heat. In fact, 40 billion BTU are produced in this way by the fission of a single pound of U-235. This may be compared with approximately 12,000 BTU obtained from burning a pound of coal. In all piles constructed so far this energy is merely dissipated in the coolant. The aim of a power pile design will be to convert the heat to useful work. Some of the difficulties in the way are discussed in the next section.

II. Engineering Difficulties For Power Piles.

The feature of a pile which dominates almost all the practical power aspects is that it must be surrounded by an enormously massive shield, as a radiation protection. In order to see the effect of the shield let us consider the power levels which are of practical interest. Table I lists the levels needed for various applications, ranging from the ordinary household electric iron up to the estimated total world consumption of power. Now the radioactivity and neutron density within a pile producing heat at only 1 Kw is already so large as to require shielding equivalent to two

FIGURE 3.—Pictured here is the Plutonium Plant of the Atomic Energy Commission at Hanford, Washington. The plant has changed in the three years since this picture was taken, but the AEC has not released any more recent pictures which would reveal changes.

feet of concrete. At the 100-Kw level, competent authorities have estimated that the minimum weight of a nuclear-powered automobile, according to present pile design possibilities, would be 50 tons.

Apparently, small piles are too heavy and too expensive to have many practical power uses, although they will always be valuable as neutron sources for research. The most important power piles will probably be built for the range from a few thousand kilowatts upwards. Practical power piles should also be designed to run at very high temperatures. It has been estimated that to construct a 75,000-Kw nuclear plant will require a capital outlay of at least $25,000,000. Such an expenditure requires that the efficiency of the plant after completion be as high as possible. And any thermal engine requires high operating temperatures to reach good efficiency.

Such economic factors alone will probably force high temperatures for piles in the range above 10,000 Kw. In the 5000 to 10,000-Kw range the potential uses as shown in Table I are chiefly in heavy propulsion, where mobility may be more important than economy. But the only apparent means of greatly reducing the weight for a given power is to find a way to decrease the size of the pile proper. The perimeter of the shield will then be smaller and so the weight will be less even though thickness is the same. However, if this reduced size for the same power can be achieved—and methods for doing this are mentioned later—there will necessarily be more BTU per second generated in each cubic foot of the pile, and the pile will run at a much higher temperature unless the efficiency of the coolant can be correspondingly increased.

Therefore, one of the central engineering problems of the practical power pile is to achieve rapid heat transfer from the uranium at high temperature to the coolant fluid, across restricted areas. Most of the suggested power pile designs which have been published contemplate that the circulating fluid will issue from the pile at something like 1200° F., which is about 650° C. or red heat. The uranium would be considerably hotter yet. Coolants suggested have ranged from high-pressure helium gas to liquid metals such as sodium or lead. These working fluids would then

TABLE I. Estimated power levels required for certain practical uses. For the mobile plants an estimate is given of the weight of typical present boilers and fuel loads which would be replaced by a nuclear pile.

Use	*Estimated Power Level*	*Estimated Weight Replaceable by a Pile*
Electric iron	1 Kw	
Automobile	100 Kw	
Submarine	5,000 Kw	630 tons
Locomotive	6,000 Kw	250 tons
Large airplane	9,000 Kw	95 tons
Electric power for a small city	100,000 Kw	
Large battleship or carrier	150,000 Kw	12,000 tons
City of a millon population	1,000,000 Kw	
Total U. S. electric power	50,000,000 Kw	
Total U. S. power	900 million Kw	
Total world power	2600 million Kw	

be led, probably through an intermediate heat exchanger, to a suitable engine for converting their heat into useful work. The closed-cycle gas turbine has been suggested by some designers. This engine then drives a generator which produces electric power as the final output of the nuclear plant. Such a system is shown schematically in Figure 4.

A number of other detailed problems are now listed, so as to indicate the kind of engineering jobs ahead. A few of these are fresh results of the shift to high temperatures, but most of them were partially solved in building the present piles—they have just been aggravated by the new emphasis on practical power.

(1) Neither the moderator, the coolant, nor any other material within the pile, such as the cans for the uranium, the coolant pipes or the process instructions, can contain any elements which are strong neutron-capturers. Otherwise, the chain reaction will be choked off. Table II lists the relative ability to absorb neutrons for various substances. Many of the common structural materials, such as steel, are immediately ruled out on this score. The requirements with respect to some elements are so extreme as to

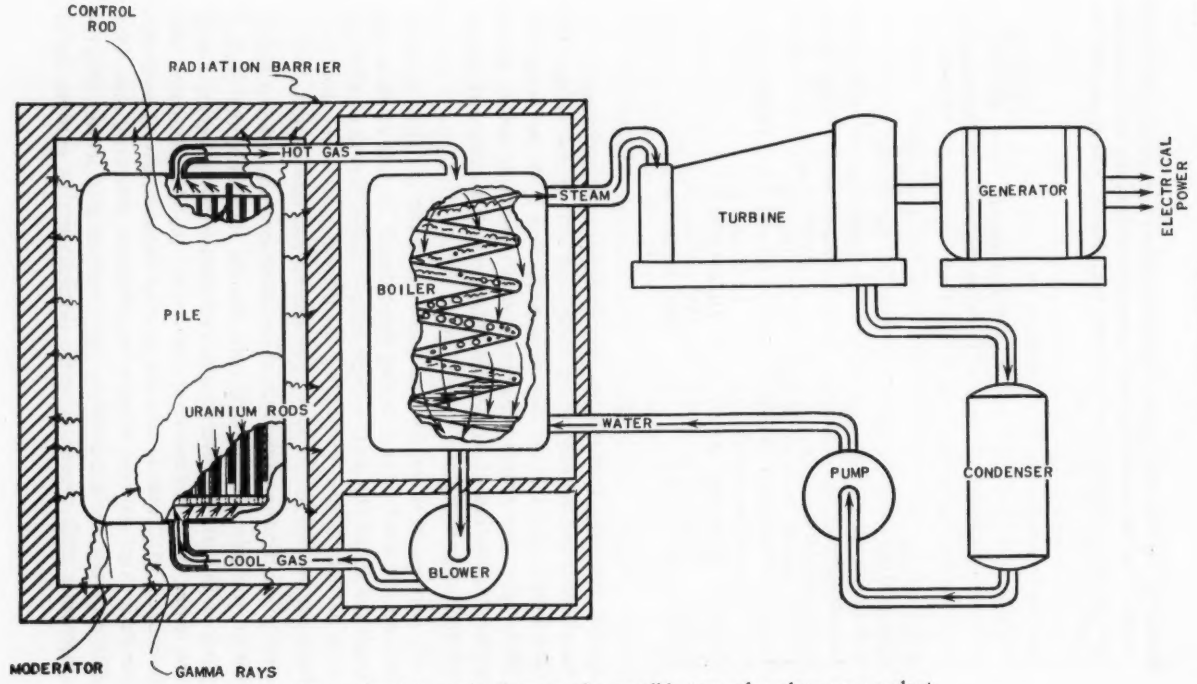

FIGURE 4.—Schematic diagram of a possible type of nuclear power plant.

TABLE II. The relative slow neutron absorbing powers for various elements. Such points as the usefulness of cadmium and boron for control rods, and the unsuitability of mercury as a coolant, are evident from such a table. (These values are from unclassified sources, based mainly on prewar measurements, and therefore are not necessarily the most modern results.)

Element	*Relative Absorbing Power*
Hydrogen	0.33
Deuterium	0.03
Helium	0.00
Lithium	66.0
Beryllium	0.01
Boron	710.
Carbon	0.0048
Nitrogen	1.2
Oxygen	0.00022
Fluorine	0.01
Sodium	0.4
Magnesium	0.31
Aluminum	0.4
Silicon	0.24
Argon	1.0
Vanadium	5.
Chromium	2.5
Iron	2.1
Nickel	6.
Copper	4.3
Silver	58.
Cadmium	2950.
Mercury	400.

FIGURE 4.—Schematic diagram of a possible type of nuclear power plant.

necessitate setting up new industries for producing materials of the necessary purity. Thus, less than one part in a million of boron can be allowed in a pile.

(2) From this restricted list, materials must be selected for all the close tolerance locations which are not subject to high-temperature creep or weakening.

(3) Materials exposed to the coolant must not react with it to corrode at high temperatures.

(4) Film formation of any kind on inner surfaces would decrease the vital heat transfer rates.

(5) The coolant (unless very carefully chosen), or impurities therein, may be made radioactive while passing through the pile. This will make the pumping machinery radioactive, and in case of trouble impossible to repair except by remote control. For this reason, the pumps and the external lines of the coolant will probably require some radiation shielding also. To decrease the amount of shielded space and to protect the large power turbines from radio-activity, the latter should probably be operated by a secondary thermal fluid, such as high pressure steam or mercury. This fluid may receive its heat from the primary coolant through a heat exchanger impervious to radioactivity.

(6) The intense neutron flux in the pile almost precludes the use of any organic materials, since the neutron bombardment will disrupt the bonds of organic molecules. Similarly, radioactivity in the coolant will break down lubricating greases in the pumps. Many inorganic liquids would also be dissociated if used as coolants.

(7) In the same fashion the strength and other physical properties of many structural materials are altered under intense neutron bombardment. This problem did not prove insurmountable in previous piles but it may be especially bad when the materials are simultaneously exposed to high temperatures.

(8) The cans for the uranium must remain absolutely gastight at the elevated temperature, since many of the heavy fission fragments form highly radioactive gases, which must not be allowed to get out into the coolant lines.

(9) Some of these heavy fission fragments, the energy of which is the source of the pile's heat, are themselves strong neutron-capturers. If their concentration is allowed to grow too high they will "poison the pile" by choking off the chain reaction. Therefore, provision must be made for periodically replacing the partly used uranium cans by fresh ones, all by remote control since everything is "hot," both thermally and radiologically.

(10) It must be remembered that though these physical problems of heat transfer and properties of materials would seem to fix the design constants of the pile in many important respects, there is always one over-riding consideration—the neutron economy. Are the dimensions and materials such as to permit

the multiplication factor to reach 1.001? Pile design represents a continuous juggling and compromising of all the interrelated constants which affect a slow neutron's chances of entering a U-235 nucleus.

Enough has probably been said by now to show that the problems, though very difficult and intricately interwined in the total pile design, are not fundamentally insoluble. A number of the engineering questions should benefit from research now being carried on by materials contractors and metallurgical and chemical contractors of the Office of Naval Research. Everyone believes that good power piles will be built sooner or later. For the stationary power plants the scale of development will doubtless depend chiefly on economic factors. Some of the relevant data are discussed in the next section.

III. Economic Aspects Of Nuclear Power.

On two primary questions depends the degree of future use of stationary nuclear power plants:

(1) Is there enough uranium, or other fissionable material, in the world to support a widespread nuclear power development?

(2) How does the cost of nuclear power compare with that of other sources?

In regard to the first question, the total relative abundance of uranium in the earth's crust can probably be estimated with sufficient accuracy from astronomical and geological data. A reasonable figure is probably four parts per million. This makes uranium at least as abundant as tungsten, cadmium, and mercury, and, possibly comparable to tin, zinc, or lead. By simple arithmetic the above figure yields ten trillion tons as the total uranium content of the Earth's crust down to a depth of three miles.

However, such figures are almost meaningless since the occurrence of uranium ores, though widespread, is almost always in very low concentration. Mineral deposits are not usually considered of commercial value unless they run over 1%. The total world uranium ore deposits of more than 1% concentration *known* in *1941* are stated (ref. 30) to have contained about 50,000 tons of metal. Now from the value for the heat release of fissioning U-235 quoted previously, one may calculate that if this heat is

turned into electric power at 10% efficiency, the fission of one pound per day of U-235 will produce about 50,000 Kw. Table I indicates that the fission of about two pounds per day of U-235 would supply the electric power needs of a small city. If one now calculates the uranium needed to satisfy the total electric power demand of the United States, (converting from U-235 tonnage to natural uranium tonnage by the factor 140), the result is about 25,000 tons of uranium per year.

We thus come to the interesting result that the total known ore deposits running over 1% throughout the world, if all available immediately, would be sufficient to power the United States electric industry for one or two years. Uranium is thus a power resource to be reckoned with, although it does not yet belong with the giants—coal, water, and oil.

However, the story is not finished. A recent report of the Atomic Energy Commission to Congress stressed the research now underway on the "breeder" type of pile. To understand this, let us return to Figure 2 and the several fates of the average neutrons per fission. Suppose the escape loss and the moderator and impurity capture losses could be so reduced that it could be arranged for more neutrons per fission to be captured by U-238 than the number, 1.000, which must be captured by U-235 to create the next generation. Then, in addition to the constant power output, the pile would be creating more plutonium per second than the amount of U-235 "burned up." This means that even after all the U-235 in the pile was used the reaction could, in principle, continue to run on the plutonium content and even continue manufacturing more plutonium faster than it used it.

Manifestly, if a power pile can be developed in which the losses are so low that it becomes also a breeder, then *all* of the natural uranium becomes available for power production, and not merely the U-235 fraction. Thus at one step the world's nuclear power resources would be multiplied by 140, and their probable period of usefulness increased from some years to some hundreds of years at our present level of power consumption.

If you grant the completion of the development just described, a further factor of 3 or 4 could be obtained by loading cans of thorium metal into a breeder power pile. Neutron capture by

thorium converts it into a rare uranium isotope, U-233, which is also fissionable—the same as U-235 and plutonium. In this manner the world's thorium resources could be used for power. Thorium is several times as plentiful as uranium and occurs in rather higher concentrations.

Thus, although very many engineering problems remain to be solved, the world fission power resources seem to be *potentially* comparable in extent to the oil resources. It then becomes worthwhile to consider nuclear costs on a competitive basis with other power plants. It must be said that enough data do not exist at the present time to make a firm estimate of the installation and operating costs of a nuclear power plant. After all, the first power pile has yet to be reported! The best that can be done is to make some guesses.

Certainly the fact that one pound of U-235 produces as much heat as approximately 1500 tons of coal does not mean that nuclear-electric power is cheaper than coal-steam-electric power. It is well known that the fuel costs are not the major item of an electric company's expenses. One must compare the actual cost per kw-hr at the outgoing bus-bar including all features of plant operation and plant cost in each case. It is assumed that distribution costs would be the same for a large plant of each type.

Although it is impossible to arrive at accurate cost figures at present, *such figures as have been published make it seem unlikely that uranium will immediately compete with coal in the general market.* However, for such special purposes as a standby to make a power network independent of coal shipments in an emergency, it looks quite possible that work on a few large nuclear plants may be begun during the next ten years.

IV. Military Mobile Systems.

The preceding section has shown that the economic motive for the development of stationary nuclear plants does not look very powerful just yet. However, the National Military Establishment would have many uses in propulsion systems for *lightweight* piles in the 5000 to 150,000-Kw range, if they could be developed. In many cases, the military urgency might outweigh considerations

of operating costs and provide an extremely strong incentive for the development.

The crucial factor in all mobile fission plants is, of course, the weight of the shielding. Table I gives some estimates of the maximum weights which could be devoted to a pile and its shielding in the largest operating types of present-day vehicles. In each case the greatest fraction of this weight would be made available by the elimination of the vehicle's fuel tanks.

There is hardly any *published* information which gives much light on the possibility of meeting these weight specifications aside from the prewar values of the fundamental nuclear constants. One small item is the previously mentioned estimate of 50 tons for a 100-Kw pile and another is the statement by several good authorities that it should be quite possible to run a battleship from a nuclear plant. Between two such widely separated datum points interpolation would be valueless, but it does seem possible that hard work may produce a nuclear-powered submarine within a reasonable time. The locomotive and the airplane look much harder.

There is a possibility that a new development may reduce the weights. This is the so-called "fast" pile, announced from Los Alamos a few months ago. If one stocked the pile with highly enriched uranium and little moderator, then the U-238 and moderator capture losses would be greatly decreased. But this would permit cutting down the dimensions of the pile without decrease in power level, because there would be sufficient neutrons to spare so that a larger escape loss could be tolerated. Use of this enrichment technique to decrease the size of the pile has already been practiced to a certain extent, but as the percentage of fissionable material present is increased, and the amount of moderator decreased, the control of the pile becomes more delicate.

Another aspect of the mobile nuclear power system has not yet been mentioned, that is the poisoning problem. Replacement of fuel elements might be necessary periodically in order to eliminate fission products which were stopping the reaction by swallowing too many neutrons. If this replacement time turned out to be a matter of days instead of months or years it might be very inconvenient.

If this difficulty exists, it should be greatly alleviated in the fast pile. When less moderating material is used the neutrons are not slowed down as much as in the "slow" piles. They will pass by any potential absorbers so quickly that their probability of being caught is much smaller. The fission products, therefore, will no longer have an important poisoning effect.

V. Future New Nuclear Sources.

No survey of the outlook for nuclear power could be complete without pointing out that fission releases only a very small part of the total energy of a uranium or plutonium nucleus. There is yet no reason to cease hoping for ways to release a much larger part of the energy of *any* nucleus—not just the somewhat rare uranium. Although no practical method is yet in sight, constant gathering of all possible facts about the structure and properties of all nuclei can conceivably strike paydirt within the next few years. The Office of Naval Research is now supporting the efforts of some 60 contractors in an aggressive program for discovering new fundamental facts throughout the field of nuclear physics.

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MILITARY CATAPULTS—THEIR HISTORY AND DEVELOPMENT.

By J. E. Scholl.

The Author was graduated from the University of Kentucky in 1934 with a B. S. in M. E.; and the George Washington University in 1943 with a B. S. in C. E. After having been associated with the General Electric Company, Schenectady, New York, and Chas. W. Cole, Consulting Engineers, South Bend, Indiana, he entered the Federal Service in 1939. Following eight months of training at the Norfolk Naval Shipyard, Portsmouth, Virginia, on the intricate construction of Naval Vessels, he was transferred to the Bureau of Aeronautics, Navy Department, Washington, D. C., in 1940, where he is now engaged in duties related to the development, installation and maintenance of catapults and launching equipment.

The story of the evolution of catapults is almost as ancient as history itself. Catapults were first used by the Romans, Greeks, and Phoenicians as instruments of war to hurl stones and blazing missiles into besieged cities. The early stone sling, the crossbow, and the great bow are catapulting devices which directly utilized man's energy to hurl missiles in one form or another at the enemy. Technological progress has eliminated these war machines, one by one, and, in more recent times has made possible equipment many times more powerful than these early crude machines. It is the recent evolution of catapults which makes the most fascinating story, principally because of the amazing energies involved. This story commenced at the turn of the century, when men were beginning seriously to design and manufacture man-carrying flying machines, after technological progress in materials had advanced to the point where such things were beginning to become possible. It is interesting to note that the development of the modern catapult has been almost contemporaneous with the development of the airplane.

The application of catapults in launching aircraft, as used today was probably the result of a number of significant events which

occurred during the early stages of the development of the airplane.

On May 6, 1896, Professor Samuel P. Langley launched a 36 pound automatically controlled airplane from a barge on the Potomac River at a location near Quantico, Virginia. Little did Professor Langley dream that almost a half century later the Navy would duplicate this feat by launching a 135 pound radio-controlled target drone airplane from a similar spring driven catapult. In this case, the Navy engineers dipped into the book of experience and brought forth an old proven arrangement which exactly fitted a special application.

In December, 1903, the Wright Brothers succeeded in launching a man-carrying plane at Kitty Hawk, North Carolina. The plane was powered by a gasoline engine and succeeded in obtaining flying speed assisted by a catapult device which was motivated by a falling weight. Seven years later, in November, 1910, Eugene Ely succeeded in flying a landplane from a temporary platform constructed on the forward deck of the U.S.S. *Birmingham* which was stationed at Hampton Roads, Virginia. This appears to have been the first demonstration of the ability of an airplane to fly from the deck of a ship. Two years later, November, 1912, Lt. T. G. Ellyson, U.S.N. who was the Navy's first aviator, successfully piloted an airplane launched from a compressed air catapult at the U. S. Navy Yard, Washington, D. C. These events proved a rising curtain to Naval aviation as it exists today.

In its modern usage, the catapult is a mechanism which accelerates an airplane from a state of rest to one of minimum flight speed. Catapults are essential for launching airplanes from a ship since they very effectively reduce the take-off runway length. It is a device consisting essentially of a source of power and means for transmitting that power to the aircraft. The choice of a source of power is necessarily limited, since it must have characteristics which allow the release of large amounts of energy during the launching period, (1 to 2 seconds). It must also be subject to very flexible control which the aircraft weight and speed characteristics demand. Although Langley used stored energy in a spring, and the Wright brothers used the energy of

a falling weight to launch their aircraft, these sources of power exhibited limitations which were not practicable in a device to launch heavy military aircraft. This point is illustrated in determining the power expended in launching a 15,000 pound airplane. If the launching velocity is 70 miles per hour, the catapult must impart approximately 2,500,000 foot pounds of energy to the aircraft. Since the efficiency of the launching device may in practice be as low as 50%, the potential energy of the power source must approach 5,000,000 foot pounds. This is equivalent to approximately 6000 horsepower when exerted in the launching time of 1.5 seconds.

Catapults have been developed under two general classifications which describe the means of transmitting energy to the airplane; the first, utilizing wire rope systems, and the second, utilizing direct drive systems. The following outline of catapulting systems, together with applicable power sources, cover most of the catapult types which have been developed by the Navy and other agencies for both military and commercial use:

1. Wire Rope Systems
    - (A) Multiple Purchase (block and tackle)
        Power Source
        - (a) Compressed Air
        - (b) Powder
        - (c) Pneumatic-hydraulic
    - (B) Single Purchase
        Power Source
        - (a) Flywheel
2. Direct Drive Systems
    - (A) Telescoping Tube
        Power Source
        - (a) Compressed Air
        - (b) Solid Propellants
        - (c) Steam
        - (d) Pneumatic-hydraulic
    - (B) Rocket Catapult
        Power Source
        - (a) Rockets
            - (1) Solid Propellants (powder)

(2) Liquid Propellants

(C) Electric
Power Source
(a) Flywheel Driven Generator

(D) Slotted Cylinder
Power Source
(a) Solid Propellants (powder)
(b) Liquid Propellants (powder)
(c) Steam

The early designers of Naval catapults, Capt. W. I. Chambers, who first conceived the idea of launching aircraft by catapult from the deck of a ship, Lt. St. Clair Smith, and Lt. H. C. Richardson, were confronted in 1911 with the problems associated with a new field of endeavor assisted by an infant aircraft industry. Regardless of monumental difficulties, the advantages to be gained by the use of seaplanes for long range scouting and gun fire spotting duties demanded a solution. Accordingly, these men adapted parts of a compressed air ammunition hoist to power the catapult. It consisted of a cylinder and ram arrangement attached to a multiple purchase wire rope system (block and tackle in reverse) which in turn pulled a car along a straight track. The first catapult to launch man-carrying military aircraft was born of this makeshift material.

The early catapult designers chose wisely in the selection of the multiple purchase wire rope system and the compressed air source of power for the first machine. It was important that the design meet the critical space limitations aboard ships. For example, the launching run of the aircraft might be as little as sixty feet. By choosing a purchase ratio of 4, 6, or 8, the engine ram stroke of 15, 10 and 7.5 feet, respectively, would be realized. The overall length of the catapult may by such a system, be varied within limits to fit a specified ship compartment. The compressed air power source was advantageously chosen because compressor equipment already available in many Naval vessels for the purposes of launching torpedoes, delivered air at pressures suitable for catapult operation. This eliminated the necessity for developing and installing separate power systems for the catapult machinery. Air operated catapults were equipped with banks of air

tanks or flasks in which air could be stored at a predetermined pressure, and released to the catapult engine through a quick opening valve. The flask recharging was accomplished over a period of several minutes to maintain a minimum of compressor capacity.

Catapults powered by compressed air exclusively were employed by the Navy for ten years after the first successful demonstration. During this period, ship designers began to acquaint themselves with the problems associated with the installation of this equipment in the ship's structure. It was not only necessary to furnish foundations to withstand the reactions attending the launching of an aircraft from a catapult, but also to modify existing vessels to provide the space required for the aircraft and aeronautical gear. In 1922, Lt. Elmer Stone, of the U. S. Coast Guard, who was on temporary duty in the Bureau of Aeronautics of the Navy Department, with Mr. C. F. Jeansen, an engineer in the Bureau of Ordnance, Navy Department, devised a catapult engine powered by gun powder. Several designs emerged from this development, one of the first of which was mounted on the top of battleship gun turrets. The turret training system was utilized to position the catapult in the most desirable direction as dictated by the relative wind direction and velocity. In another form, the powder-operated catapult engine was installed in a truss structure carrying a track. The whole rotated in azimuth on a platform. Powder-operated catapults to launch seaplanes were eventually installed on most all battleships and cruisers in the Navy.

While the early air and powder-operated catapults admirably served their purpose, the Navy was not satisfied to rest on these laurels. A development which was successfully prosecuted during the period, 1925 through 1927, utilized the energy stored in a revolving flywheel to launch the aircraft. This device consists of a flywheel, a clutch, a tapered drum, a wire rope towline, and a brake. The flywheel is easily brought up to speed, using moderate power for a short time. When the aircraft is ready for launching, the flywheel is connected to the stationary drum by means of the clutch mechanism. The transferred energy causes the drum to revolve and wind up the towline connected to the aircraft. Since

the drum is tapered, it is possible to start the towline slowly and then quickly build up speed for the launching velocity, with substantially constant acceleration. At the end of the run the flywheel is declutched, and the drum is brought to rest by use of the brake. The device demonstrated the applicability of the single purchase wire rope system; that is, one in which the catapulting mechanism is accelerated to the same speed as the aircraft. Two flywheel catapults, for launching twin float seaplanes, were built by the Navy and installed, one each, in the original U.S.S. *Saratoga* and U.S.S. *Lexington*.

A development which assisted materially in reducing the weight and maintenance of catapults was the introduction of the pneumatic-hydraulic power system. In the air catapult, the air pressure acts directly on the ram. This results in the loss of a large volume of high pressure air which is exhausted from the engine after each launching. In the pneumatic-hydraulic catapult a large accumulator of fluid is introduced between the air flask and the engine. The development of hydraulic catapults was initiated in 1934. The fluid volume is large enough, and is arranged properly with respect to the air flasks to prevent loss of air or entrance of air to the engine during firing. During each launching, the fluid is exhausted to a sump tank at atmospheric pressure from which it can be recharged into the accumulator by means of a hydraulic pump. By this process, the air in the air flasks is recompressed to the desired pressure. The equipment resulting from these developments was very successfully employed on Naval carrier vessels during World War II.

The telescoping tube catapult consists of a number of tubes or rams arranged in the form of a telescope. The outer tube is held fixed in the catapult structure, or to a foundation, and the smallest tube is attached directly to the aircraft, or to a launching car supporting the aircraft. When the actuating force (high pressure air, gas generated from solid propellants, steam or hydraulic fluid,) is injected into the after end of the outer tube, the annular areas of the inner tubes exposed to the high pressure are forced ahead to launch the aircraft. Since the tubes are moving masses which must be brought to rest after the aircraft is launched, the last portion of the stroke of each tube is designed to act as a buffer and retard the tubes to a stop.

The design of a telescoping tube catapult, similar to that described above, was undertaken by the Royal Aircraft Establishment, Great Britain, in 1922. A number of these catapults were subsequently manufactured and successfully used to launch aircraft from British vessels.

While this type of catapult is suitable for specific installations, the arrangement did not meet the U. S. Navy requirements, and conditions as effectively as other types of catapults. For this reason the U. S. Navy did not pursue the development beyond the preliminary study phase.

A novel catapult designed by the Navy in 1943 consisted essentially of a rocket, shuttle designed to slide in a set of channel tracks and a brake. The rocket was attached to the shuttle with the rocket nozzle pointing toward the rear. The aircraft was attached to the shuttle with a wire rope pendant. When the rocket was ignited, the generated thrust on the shuttle pulls the aircraft ahead to flying velocity. The brake stopped the shuttle and empty rocket at the end of the launching run. Care must be exercised in this arrangement to position and shield the aircraft from exhaust blast of the rocket. In 1944, a Piper Cub aircraft was successfully launched from a U. S. Naval vessel of the LST class with this rocket catapult.

Another version of launching aircraft with the use of rockets, utilizes the rockets by fastening them directly to the aircraft. After take-off the empty rocket cases are jettisoned by the special attachment fittings which hold them to the aircraft. This system is not strictly a catapult, as are the other devices described herein but it accomplishes the purpose of launching an aircraft in a distance much shorter than can be accomplished with the aircraft main propulsion system alone. The method is called JET ASSISTED TAKE-OFF or JATO.

The electric catapult consists essentially of an electric induction motor and a source of electric power. Instead of a motor of cylindrical configuration normally seen in the home for powering refrigerators, washing machines, etc., or in industry for driving lathes, drill presses, conveyors and other machinery, the components of the motor, the rotor and the stator, are flattened into a linear arrangement. A moving launching car about ten feet

long contains the primary copper windings normally found in the non-moving part of a motor (stator) and the stationery track-like part of the catapult, over 1300 feet long, contains the squirrel cage windings. Power is fed to the launching car's primary winding through a pickup arrangement. This power induces a current in the track's windings and develops the electromotive force to push the car down the track.

Since a considerable amount of electric power is required to operate the catapult, it was necessary to devise a system which would keep the primary source of power as small as possible within cycling limitations. The flywheel was ideally suited for the purpose. While the complete power system appears somewhat complicated because of the numerous units used, it is, in fact, quite simple. An 1,100 horsepower aircraft engine drives a d.c. generator which is wired to a d.c. motor. The d.c. motor drives a 12,000 KW alternating current generator and the flywheel on the same shaft storing the energy in the flywheel during the five to seven minute pre-launching period. The flywheel then provides energy during the three or four second launching period, by driving the a.c. generator to supply the power demand of the linear motor.

The electric catapult was developed for the Navy by the Westinghouse Electric Corporation during the period, 1943 to 1945. Being too long and heavy for shipboard use, it was not generally adopted.

A unique direct drive catapulting system, developed and utilized by the Germans during the late war, is usually described as a "slotted cylinder" catapult. It consists of a long tube or cylinder closed at one end with a narrow slot cut along the longitudinal axis at the top, a piston fitting the inside diameter of the cylinder, a tow fitting attached to the piston, a means to seal the slot from inside the cylinder, and a gas generating system. In the German catapult, the cylinder was approximately twelve inches inside diameter, and one hundred and fifty feet long. The piston looked like a large dumbell with a thin plate fin or tow bar extending several inches beyond the outside diameter of the piston. When the piston is placed into the cylinder, the tow bar fits into the slot and projects through the tube for purposes of attaching

to the aircraft. The arrangement allows the piston to be pushed down the length of the tube. In catapulting, a high pressure gas is injected behind the piston, creating the force which accelerates piston and aircraft. The trick, of course, is to keep the gas from escaping through the slot behind the piston. This is accomplished by means of a small auxiliary tube which is threaded through the piston and is of a length equal to that of the cylinder. As the piston travels forward, the tube is guided backward and upward by the piston to a position in line and just under the slot opening. The gas pressure behind the piston forces the sealing tube into the slot, thereby effectively sealing the gas within the cylinder. It is necessary also to suspend that portion of the sealing tube ahead of the piston at a position two or three inches below the slot. Copper wire hangers spaced at five foot intervals along the longitudinal axis satisfied this requirement. The shape of the slotted cylinder is maintained against internal pressure by the use of diaphragms spaced along the cylinder. In each launching, the piston is ejected from the end of the tube as a bullet from a gun. To propell the piston along the cylinder, the Germans injected the steam product of the decomposition of highly concentrated hydrogen peroxide.

The slotted cylinder catapult was employed by the Germans to launch the V-1 buzzbomb against London, and other targets during the war.

In the brief description of the various catapult systems, mention of the means of attaching the aircraft to the catapult power transmission system for launching has been omitted. In this connection, two general systems, one for seaplanes and one for ship based landplanes have been employed, each applicable to any of the catapult types which have been developed.

The early catapults which were designed to launch seaplanes, employed a launching car to hold the aircraft in proper attitude before and during a launching. Since the expensive construction of the car prohibited its expenditure overboard with each launching, it was necessary to provide a set of buffers at the end of the catapult to stop the car. The fittings on the aircraft were designed to mate with fittings on the car. When the aircraft is launched, these mated fittings release the aircraft as the car is arrested by the buffers.

The launching of carrierbased landplanes is the more fascinating of the two launching arrangements. It is commonly called "flush-deck" launching, and was developed by the Navy beginning in 1931 under the direction of Lt. L. J. Maxson and Mr. F. B. Gross, the latter still in the Bureau of Aeronautics. The inspiration for the development was probably derived from the stunt performed some twenty years before, when Ely flew a landplane from the deck of the U.S.S. *Birmingham*. By this time, however, aircraft and the aircraft carrier had advanced to a position of great importance as units in the Navy. The carrier launching system consists essentially of a shuttle with a tow fitting, a set of shuttle tracks, a wire rope bridle, and a hold back release device, in addition to the catapult engine. In the case of the wire rope systems, the catapult rope is attached to the shuttle which slides in the tracks located just beneath the deck. The tow fitting projects above the deck through a narrow slot extending the length of the track. The aircraft is centered with respect to the slot in the deck and attached to the shuttle tow fitting by means of the wire rope bridle. The hold-back release fitting is attached to the aircraft at a convenient point, and through it, to the deck. The holdback restrains the aircraft against full propeller thrust during the prelaunching period, and releases the aircraft at the proper time. At the instant the catapult is fired, the shuttle moves forward down the track, towing the aircraft along on its landing gear.

It is necessary to stop the shuttle and other moving parts of the catapult when the aircraft has reached flying velocity. Special braking devices on the catapult engine consisting of a set of buffers and an auxiliary multiple purchase rope system are furnished. The auxiliary rope system is connected to the after end of the shuttle. As the ram of the engine strikes the buffers at the end of the launching, the moving parts are brought to rest through the action of the buffers and the auxiliary rope system. The catapult engine is installed below the launching deck.

SUMMARY.

The Navy has been amply rewarded for the many years spent in catapult development. While many other weapons, including

those of the enemy, were well publicized during and after the recent war, the catapult performed its task in a more unobstrusive manner, a "silent service" if there ever was one. With each major engagement of the late war, Guadalcanal, the Battle of the Atlantic, the African Campaign, Okinawa and many others, the aircraft carrier task forces showered hordes of aircraft into the skies to support beachhead landings, and to mercilessly hammer enemy fleet units. The concentration of these attacks was directly attributed to the reliable catapult equipment provided.

On the part of the pioneers who resurrected the catapult from antiquity, and envisioned its modern usage, the catapult engineers who, step by step, perfected launching equipment and procedures, the aircraft designers who contributed the equipments and designs which prepared aircraft for catapulting, and the naval constructors who perfected the techniques of installing catapults and associated aeronautical gear in the vessels, it was a job well done!

DIGITAL COMPUTERS AND THEIR APPLICATIONS.

By Albert E. Smith and Dr. C. V. L. Smith.

I—Principles of Digital Computers.

Mechanical aids to computation have existed in various degrees of perfection ever since men have been faced with the need of carrying out calculations. These have assumed two principal forms, the continuous and the discrete: the former extend in range of complexity from the slide-rule to the differential analyzer; the latter from the abacus to the automatic sequence controlled digital computer. In common parlance, these two classes are referred to as "analogue" and "digital" computers respectively. We should not, however, consider these as the only possible types, for it is quite possible that the combination of continuous and discrete elements in a single system may prove, at least for certain purposes, to be advantageous.

In general, in continuous computers, the quantities upon which operations are to be performed are represented by physical magnitudes, called the "analogues" of these quantities, such as lengths, shaft positions, voltages, or any convenient physical entities. The computer simulates the system whose performance equations are

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to be solved, in the sense that it is so set up that its own performance is described by these same equations, and solution consists of recording the performance of the computer. There thus exist a great variety of possible continuous computers, depending upon the physical elements chosen, and the type of mathematical problem which it is desired to solve. Continuous computers are usually applicable to only restricted classes of problems: for example the differential analyzer was designed to solve systems of ordinary differential equations. As to precision, this is determined by the tolerances that can be held in constructing the components, and by the precision with which the quantities used to represent the variables can be measured. It is furthermore influenced by such things as temperature and humidity, and by the aging of components.

On the other hand, in a discrete device, the numbers are considered as written in a positional notation, such as the common decimal notation, or the binary system to be considered below, and some discrete device is used to represent each individual digit. In this way, numbers of as many decimal places as desired may be represented by adding sufficient equipment; precision therefore is a function of complication of equipment rather than of the tolerances that are held in its construction. Numbers can then be combined according to the rules of arithmetic. This is sufficient, since a limiting process, such as integration, can be approximated as closely as can be desired, by finite sequences of arithmetical operations.

A device of this type is a digital calculating machine. The simplest types in common use are the ordinary desk machines; an operator is required to insert each item of numerical data, and to indicate to the machine which operation it is to perform.

A calculation may be very extensive, requiring a large number of arithmetical operations; in the interest of speed and efficiency, it is desirable to make the process automatic as far as possible. One of the first significant steps in this direction was made by the International Business Machines Corporation in the various items of business machine equipment produced by them, in which the data are punched on cards, then automatically fed to the machine, which "reads" the cards, performs specified operations

on the data, and punches the results on cards. The sequence of operations is thus in part automatically controlled, and human operations are required, once the data cards have been punched, only to transfer sets of cards from machine to machine.

The next step is to make the whole process fully automatic, so that, once an operator has prepared the data of the problem in a suitable form, for example as holes punched in a paper tape, and provided the machine with suitable instructions to control the sequence in which operations are to be performed, the solution of the problem will proceed automatically, and the answer be recorded.

Numbers and instructions are ordinarily presented to the machine as sequences of electrical impulses. Physically they are not distinguishable, and are indifferently referred to as "words." The machine must be instructed whether to interpret a given word as a number or an instruction. Words are introduced into the machine and stored until they are needed, or they may be introduced as required. For example, it is possible to store the words representing the numbers which are the data of the problem within the machine, while the instructions are introduced one at a time. In any case, the machine is caused to read successive instructions, and to operate upon the designated numbers accordingly. The result of each operation may be stored within the machine for future use, or if it is not required further in the course of the calculation, it may be caused to actuate a device which presents an appropriate external indication.

Any machine of the type described above is called an automatically sequenced digital computer. Our analysis of the machine functions shows that such a computer must include the following components: a device by which an operator can record upon a suitable medium the numerical data and the sequencing instruction; a device which can "read" this information and transmit it to the machine; a means of storing this information within the machine; a device for performing the operations of arithmetic; a device which interprets the instructions and causes the desired arithmetical operations to be performed; a means of removing the results from the machine, and of transforming them into the conventional numerical symbolism. There is no fixed

term for the first device on the list; the others are commonly called the input, the internal storage (a "memory"), the arithmetic unit, the control, and the output. The output will ordinarily drive some sort of printing device as mentioned above. A block diagram illustrates the general plan of machine organization:

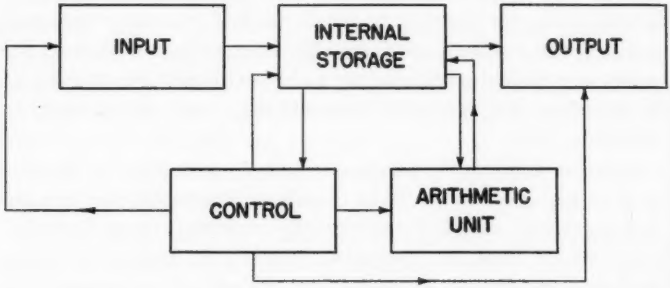


FIG. 1 FUNCTIONAL BLOCK DIAGRAM OF DIGITAL COMPUTER

Before discussing actual and proposed methods of mechanizing the blocks of the diagram, it is necessary to consider the representation of numbers. If the ordinary decimal system is used, it is necessary to have means of representing each of the digits 0, 1, 2, ...9. A single physical element which has ten distinct states is needed to represent each digit. Alternatively, some means of representing a digit by means of a group of elements having a smaller number of distinct states suffices. As a mechanical element capable of ten distinct states, a counter-wheel will serve as an example.

It is generally simpler to find an element having two distinct states; a simple example is a relay, which is either energized or not, or on Eccles-Jordon "flip-flop" circuit. The latter consists of two vacuum tubes (triodes or pentodes), and is such that just one tube can be in a conducting state at a time; the circuit possesses two stable states, according to which of the two tubes is conducting. By cascading relays or flipflops it is possible to build up devices which have ten distinct states.

The comparative simplicity of devices having two distinct states suggests the use of the binary system of notation. The common decimal system is a positional notation based on powers of ten, that is, the number 2456 means

$$2 \times 1000 + 4 \times 100 + 5 \times 10 + 6$$

or, if we recall that $10^0 = 1$,

$$2 \times 10^3 + 4 \times 10^2 + 5 \times 10^1 + 6 \times 10^0$$

Binary notation is a similar positional scheme based upon powers of 2, and thus requires just two digits, zero and unity. Thus reading from right to left, the first column is units, the second 2's, the third 4's, the fourth 8's, etc. For example, the number 115 can be written as a sum of powers of two as follows:

$$115 = 64 + 32 + 16 + 2 + 1$$
$$1 \times 2^6 + 1 \times 2^5 + 1 \times 2^4 + 0 \times 2^3 +$$
$$0 \times 2^2 + 1 \times 2^1 + 1 \times 2^0$$

Using binary notation this becomes, remembering that 2^0 is 1:

1110011.

A very simple method of converting from decimal to binary is by successive division by two:

115	
57.5	1
28.5	1
14.0	0
7.0	0
3.5	1
1.5	1
.5	1

The result of the first division is 57.5. Note the fractional part .5, and record a 1; discard the fractional part and divide 57 by 2; again a fractional result is obtained, 28.5. Again a 1 is recorded, the .5 discarded, and the process continued. When a number without fractional part is obtained, a zero is recorded, and the division process resumed. The binary equivalent is then the succession of ones and zeros recorded, read from the bottom up (that is: 115 = 1110011 in the binary system). Similar simple schemes have been devised to convert integers from binary to decimal notation, and of converting fractional numbers from one scale of notation to the other.

Binary arithmetic is particularly simple. The addition tables are $0 + 0 = 0$, $1 + 0 = 1$, $1 + 1 = 10$ (the binary representation of 2), and $1 + 1 + 1 = 11$. For example, consider the addition of 1101 (13) to 1011 (11)

```
  1101
  1011
-------
 11000
```

Beginning at the right, $1 + 1$ gives 10, and hence we record a zero and carry a 1; the rest of the process is similar. The result is equivalent to 24 as it should be $(13 + 11)$. Binary multiplication depends on the multiplication table

$$1 \times 0 = 0$$
$$1 \times 1 = 1$$

Thus

```
    1101
    1011
   -----
    1101
   1101
  11010
--------
 10001111
```

the result is seen to be equivalent to the decimal number 143.

Subtraction and division are similarly very readily accomplished.

Instead of directly expressing a number in the binary notation, it is possible to retain the decimal system but to represent each decimal digit by its binary equivalant. This mixed system is called "binary coded decimal". Each decimal digit requires as many as four binary digits for its representation. Thus an n - place decimal number will be represented by 4n characters, in binary coded decimal system while if translated into the binary system, approximately 3.3n characters will be required, so that the binary representation of numbers by the machine is slightly more efficient than coded decimal.

The organization of a typical digital computer has already been discussed, and an over-all block diagram shown (Fig. 1). At this

point it is of interest to indicate briefly some of the means by which the desired functions can be realized; in each case a variety of means have been used or suggested.

The input can be effected by means of punched paper tape or IBM cards if the rate of input is not to be high, and these methods have been employed successfully. For higher speed input, work is being done on the use of magnetic tape, and it is expected that several machines now being built will use this means. A tape is coated with material that can be magnetized by the application of a magnetic field, and the words "written" on the tape as sequences of magnetized and unmagnetized spots. This information can be put on the tape from a typewriter-type keyboard. To read the information into the machine the tape passes a gap in a ring shaped core of magnetic material; this core carries a winding, so that, when a magnetized spot passes the gap, part of the magnetic field of the spot passes through the core, and induces a voltage across the terminals of the winding.

Of course the paper tape, IBM cards, or magnetic tape must be prepared manually before being read into the machine.

Output can be accomplished by much the same methods as input. In many cases, the results of the computation must ultimately be presented in the form of ordinary numbers in our common decimal system of notation. This is accomplished by causing the output medium, whether paper tape or magnetic tape, to be read by a device which drives an automatic typewriter.

The internal "memory" or "storage" is particularly important in fast machines. Words which are read into the machine must be stored in such a way that they may be made available to the arithmetic unit with very small delay. Thus, two important characteristics of the high-speed storage are the number of words it will hold, and the time required to get a desired word out of the storage when needed; these characteristics have been named the "capacity" and the "access time" of the storage. It is important that a word put into the stroage be put in a definite location, so that it can be recovered at will. The specification of this location is commonly referred to as the "address" of the word.

A great variety of storage devices are possible, and the one chosen is governed by a variety of considerations such as desired

capacity and access time—the last being necessarily determined by the desired speed of computation; cost and complexity of equipment are also very important factors.

A fast but expensive method of storage is a set of flip-flop circuits, (a "flip-flop register") one for each binary digit in the word to be stored. The flip-flop is a two-tube circuit as shown in Fig. 2

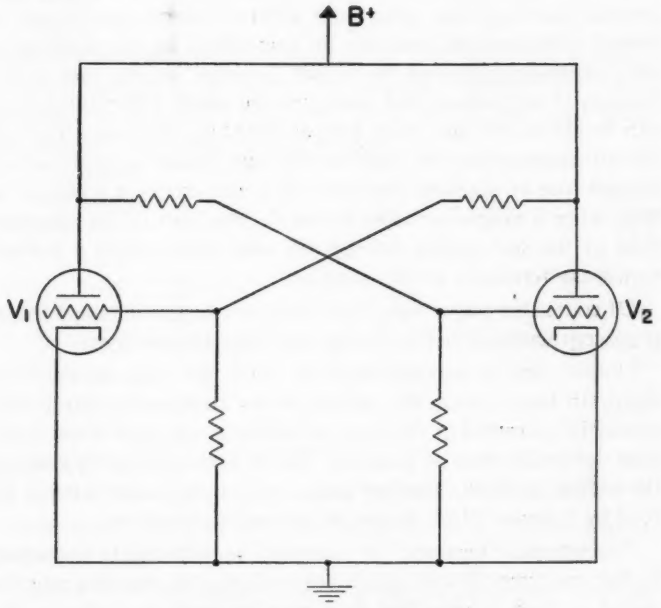


FIG. 2 FLIP FLOP CIRCUIT

It is capable of two stable states; either with V_1 conducting and V_2 off, or vice versa. It can be driven from one stable state to the other by applying an appropriate signal to the grids of the tubes. The flip-flop is seen to be very well adapted to the binary scheme of representation of numbers, since one stable state can be used to represent 1 and the other 0. Numbers can obviously be read in to a bank of flip-flops very rapidly, and the read out can be accomplished rapidly also by sensing, for example, the plate

voltages of the tubes. However, storage of a large number of words by means of flip-flop is not practicable, because of the very large number of tubes required and the switching problems involved. Flip-flop registers will ordinarily be used only to store a restricted number of words which must be made available with very small access time.

Another method of storage that can be made to have short access time is the so-called electrostatic type. Here the storage mechanism consists of charging small portions of a dielectric plate by means of bombardment by a beam of electrons. Various methods of doing this have been suggested. It has recently been reported that Professor F. C. Williams of the University of Manchester in England has succeeded in adapting commercial cathode ray tubes for this type of storage. Others have tried to devise special tubes. One American project is the "Selectron" being developed by the Radio Corporation of America.

A storage device that is receiving a great deal of attention, and which will be incorporated in several machines presently being constructed, is the acoustic delay line. In principle this is very simple (Fig. 3)

A tube is filled with mercury, and closed at each end with a piece of quartz crystal. If an electrical impulse is applied to crystal A, a mechanical vibration will be set up, which will be transmitted through the mercury to crystal B, which it will set in vibration. This will cause a voltage to appear across crystal B. The velocity of propogation of the mechanical vibrations through this mercury is comparatively slow: the time required for a pulse of energy to travel the length of a tube 4.7 ft. long at room temperature is approximately 1000 microseconds. Thus, a train of pulses arriving at A will appear at a later time at B. If the pulses of voltage obtained from B are now amplified and applied again at A, it is evident that they can be caused to circulate indefinitely through the mercury line and the external circuit. Actually it is necessary to be careful to recreate the pulses in the electrical circuit between B and A, as they become widened or "smeared out" during transmission from A to B.

Ordinarily, the mercury delay line is made long enough so that several "words" are in transit through the mercury at once. Hence,

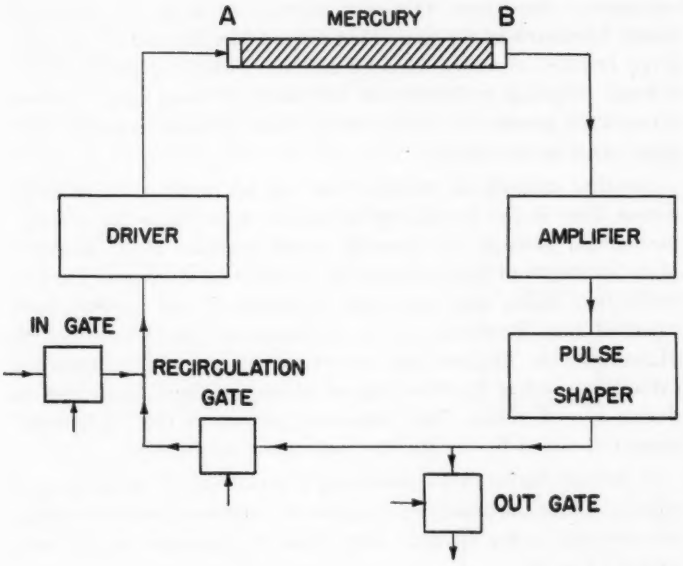


FIG. 3 DELAY LINE MEMORY UNIT

if a specified "word" is to be read out of storage, it is necessary to wait for it to emerge from end B of the delay line, which can be done by keeping track of the timing. At the proper instant an out "gate" is opened so that the required word can be read out. The simplest example of a "gate" is a tetrode, with the output from B applied to one grid, while the other is kept at low potential to prevent conduction: the pulse will be transmitted only if the potential of the second grid is raised to an appropriate level. Clearly then the average access time will be half the time necessary for the acoustic vibration to travel from A to B. It may be asked why under these circumstances it would not be better to have a great number of short delay lines rather than a smaller number of long ones, as in this way the average access time could be greatly reduced. The reason for not adopting this expedient is that it would greatly multiply the amount of equipment needed,

as the number of vacuum tubes in the recirculating system between B and A is essentially independent of the length of the line.

Still another means of storage, with access time reasonably small though larger than the delay line, is to store the binary digits as magnetized spots on the surface of a rapidly rotating magnetic disc or drum. This method has the advantage that the spots are permanent until it is desired to erase them, and thus temporary failure in power supply or in the functioning of this machine does not cause the contents of the internal memory to be lost.

The arithmetic unit is the part of the machine where the actual operations of arithmetic are performed on numbers. It can be constructed in a great variety of forms. It may use electromechanical counters, relays, or vacuum tube circuits for performing the operations.

Addition can, for example, be accomplished by either a "serial" or a "parallel" device. Consider the addition of the binary numbers 101. and 11. In a serial adder, the digits of the two numbers which are in the column of lowest value are first added; if there is any carry, it is delayed until the digits in the next positional column arrive, when it is added to their sum, and so on. Thus, first the device adds 1 and 1. This gives a 0 output and a carry of 1. The 1 is held until the 0 and 1 in the second column arrive, when it is added to them. This gives a 0 and a carry of 1. The carry is held until the third column digits, 1 and 0, arrive. Addition of the carry to these results is a 0 and a carry of 1 which is delayed until the next pulse time, and then added to the fourth column digits which are both zeros, thus giving an output of 1. Thus the sum of 1000 is obtained.

In a parallel device, the digits in corresponding columns are added simultaneously, and the carries subsequently dealt with. Thus, we first obtain 110 with a carry to be added into the second column. Addition of this gives 100 with a carry into the highest order column; finally addition of this carry gives 1000 as the sum.

As an illustration of the circuitry by means of which addition may be accomplished, consider the case of a serial adder. This can be built up of two "half-adders", which are illustrated in Fig. 4.

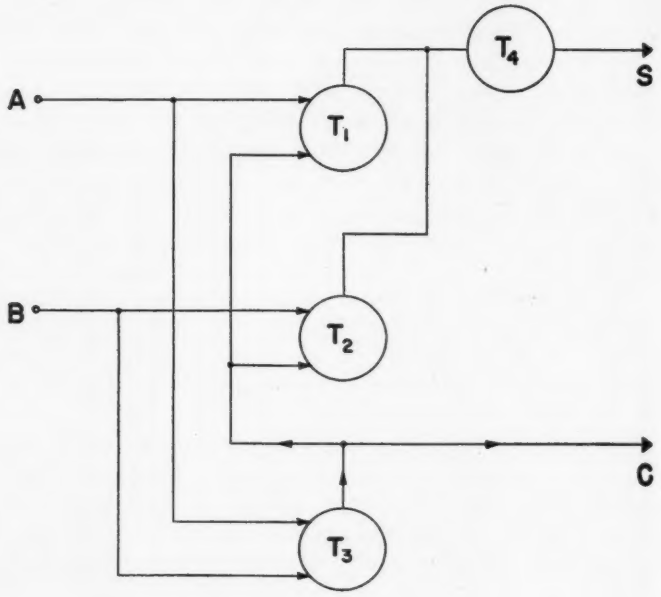
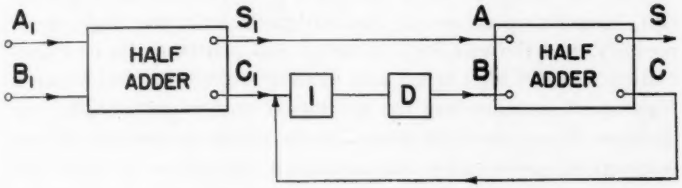


FIG. 4 HALF ADDER

Tubes T_1, T_2, T_3 are tetrodes, T_4 a triode. Inputs A and B are applied respectively to the two grids of T_3, and to one grid each of T_1 and T_2. The second grids of T_1 and T_2 receive their inputs from the plate circuit of T_3. In the absence of any inputs, T_1, T_2 and T_3 are biased below cut-off, and T_4 is conducting. T_3 does not conduct unless pulses are applied to both grids. Suppose now a positive pulse arrives at A and no pulse at B, then T_1 will be caused to conduct during the pulse, and its output to T_4 will be a negative pulse, which will be inverted by T_4. Hence 1 at A and 0 at B will give 1 at S (sum), while clearly 0 at A and 1 at B will give the same result. If positive pulses appear at both A and B simultaneously, T_3 will conduct, and the negative pulse from its plate circuit will inhibit tubes T_1 and T_2, so that no pulse is delivered to T_4 and hence to S. Thus $1 + 1$ gives a 0 at S, and a

negative pulse at C (carry). A serial adder can now be constructed; for consider the block diagram, Fig. 5.



I — INVERTER

D — ONE PULSE TIME DELAY

FIG. 5 SERIAL ADDER BLOCK DIAGRAM

The purpose of the inverter is merely to change the output from C from a negative to a positive pulse. The delay is introduced so that the carry pulse will be added in the succeeding column position. Consider the sum 101 + 111. The 1's in the unit's position first appear at A_1 and B_1. Hence 0 appears at S_1, and a negative pulse at C_1. This is inverted and delayed one pulse time. There being no input to A or B, no output (a zero) appears at S. In the next pulse time, 0 appears at A_1, and 1 at B_1; hence 1 appears at A, and the delayed carry from previous pulse time appears at B. Hence no output (0) appears at S, and a negative pulse at C, which is inverted and delivered through the inverter to the delay line. In the third pulse time, 1 appears at both A_1 and B_1, hence 0 at S_1, and a carry at C_1. At A, no pulse appears, since the output from S_1, is zero, but the delayed carry from the preceeding pulse time appears at B. Hence 1 appears at S. In the next pulse time, no inputs appear at A_1 and B_1, but the delayed carry appears at B and hence a 1 appears at S. Hence the result of the operation delivered from S is 1100 which is seen to be correct. Other serial adders can be built in a variety of ways. Parallel adders can be built up of flip-flops and gate tubes. No illustration will be given here.

Multiplication can be performed by successive shifting and adding and division by successive shifting and subtracting.

The control keeps the various units of a computer in synchronism, controls operation of the arithmetic unit, the high speed memory, and the auxiliary memory, and causes shifts of information between high speed and auxiliary memories and between high speed memory and the arithmetic unit as required by the problem. The control can automatically recognize and act on such instructions as to carry out arithmetic operations, to move the auxiliary memory medium, to read, erase, or write information on the auxiliary memory, to extract certain parts of a word, and to choose one of two alternate sequences depending on which of two numbers is the greater.

In order for the control to exercise its function, high speed switching means are needed, in such operations as selecting memory locations routing information, and performing arithmetic processes. One of the most promising devices is the resistance or diode matrix. The binary number for example, representing an address in the memory can be set up on a flip-flop register and used to select the gate to that particular memory location. The operation will be explained by use of a four position resistance matrix (Fig. 6). The resistors attached to E, F, a etc. are of very high resistance.

Two flip-flops can be used to select any one of four outputs. Suppose the left hand tube of each flip-flop is conducting, then wire # 1 is selected as can be seen from the following description. Points a and b of wire 2 and 4 will be at low potential because left side of ff 1 is conducting, and points c and d will be low potential points due to conduction of left side of ff 2. Points E and F will tend to be at B^+ due to non-conduction of the right side of ff 1 and similarly for G and H due to non-conduction of right side of ff2. Since wire 1 is attached through resistors at E and G to the non-conducting halves only of the flip-flops, wire-1 has the high potential and is the selected wire. R_1 will of course draw a little current through such paths as that through E to line 6, through F to line 3, through C to line 7, and to the left half of ff2. To minimize this loading effect, crystal or vacuum tube diodes are now being used. It can be seen by similar reason-

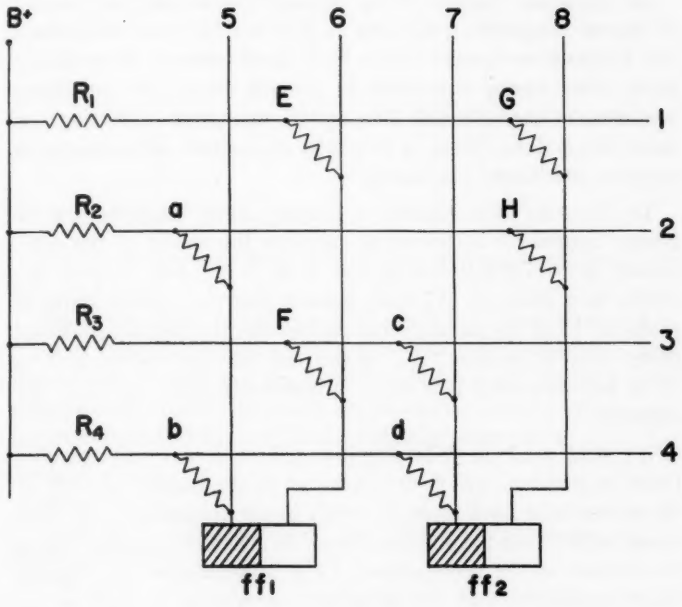


FIG. 6 RESISTANCE MATRIX SWITCH

ing that if the left half of ff1 and right half of ff2 are conducting, line 3 will be selected. Three flip-flops attached to a properly designed matrix can be used to select any one of 8 wires, or n flip-flops will select any one of 2^n wires.

One way to design the control function is to have the control start at a point in the high speed memory and execute sequentially the instructions found in successive memory locations. That is, the instruction stored in memory location 300, for example, is transferred to the control and the control causes this instruction to be obeyed by the computer. Then the instruction in location 301 is similarly carried out and so on in sequence. According to another school of thought, each instruction should include the memory location of the next instruction to be carried out.

An important feature of the dynamic sequencing and control of several computers now being built is the fact that instructions and numbers are stored in the high speed memory in a similar form. This makes it possible to provide means for modifying instructions stored in the memory by arithmetic operations on these instructions. Thus, a flexibility of control not available in previous machines is achieved.

To illustrate this feature, a simple coding example will be given. Suppose it is desired to calculate the vaules of the polynomial $4x^2 + 7x + 9$ for $x = 2, 3, 4, 5 \ldots$ and to store the results in a memory. We shall assume that the control starts at position "1" in the memory and carries out the instruction found there, then to position "2", carries out that instruction and so on in sequence until it reaches an instruction that breaks off the sequence.

We shall need the following instructions A(m)—add number found at position "m" in the memory* to the number already in the accumulator, and leave the result in the accumulator. A(503) would mean: add the number found in memory location 503 to the number in the accumulator. (The accumulator is a register in the arithmetic unit for temporary storage of a result of an arithmetic process.)

M(m)—multiply the number in memory position "m" by the number in the accumulator and leave result in the accumulator.

C(m)—transfer the number in the accumulator to memory position "m", leaving "0" in the accumulator.

T(m)—instead of continuing in sequence, perform the instruction found in memory position "m" and proceed from there in sequence.

The instructions and numbers to be used in this problem, let us say, will be stored on magnetic wire and transferred to the internal memory positions in sequence.

We store the following constants and coefficients in memory locations 31 to 35.

*When a number is read from the memory into the accumulator for addition, multiplication, etc., the number is not erased from the memory. Reading a number into the memory erases what is already there.

TABLE I.

Memory Locations	*Number Stored*
31	x
	(2 Stored at start of problem)
32	1
33	4
34	7
35	9

We rewrite the problem, putting the polynomial into the form $x(4x+7)+9$.

The instructions required are shown below. Instructions 1 to 6 are required to evaluate the polynomial for a value of x and instructions 7 to 13 are required to modify x and the first 6 instructions and return the control to memory location 1 to evaluate the polynomial for the next and succeeding values of x.

TABLE II.

Memory Location of Instruction	*Instruction*	*Result of Instruction*
1	A 31	Add number in memory location 31 into accumulator. x(2 in first case) now in accumulator; since, from table I, x was in location 31
2	M 33	Multiply number in accumulator, which is x as a result of instruction no. 1, by number in memory location 33, which from table I is seen to be 4. Result (4x) now in accumulator.
3	A 34	$4x+7$ now in accumulator.
4	M 31	$x(4x+7)$ now in accumulator.
5	A 35	$x(4x+7)+9$ now in accumulator. The value of polynomial for a given value of x ($x=2$ in first case) is in accumulator.
6	C 50	$x(4x+7)+9$, for $x=2$ in first case, now in memory location 50, accumulator cleared.

TABLE II—Continued.

Memory Location of Instruction	*Instruction*	*Result of Instruction*
7	A 31	x in accumulator (x = 2 in first case).
8	A 32	1 is added to x, x now will = 3 for the second case, x = 4 for third case, etc.
9	C 31	Modified x back in memory location 31, accumulator cleared.
10	A 6	Instruction in memory location 6, which is C 50, now in accumulator for modification.
11	A 32	1 is added to instruction. (Instruction will read C 51 to point out memory location for result when x = 3, C 52 when x = 4, etc.)
12	C 6	Instruction C 51 now back in memory location 6, for the next case C 52 will be in location 6, etc.
13	T 1	Control now returned to memory location, 1, (polynomial will be evaluated for x = 3 in second case, result stored in 51, instructions modified, control then returned again to location 1, then polynomial evaluated x = 4, result stored in 52, etc.)

In actual practice, the information like that in tables I and II would be typed on magnetic or paper tape and inserted into the computer. A button would be punched and this information automatically transferred to the internal memory. When the transfer was completed, the control would start at the instruction found in memory location 1 and proceed to carry out this instruction and those found in successive memory location.

As coded above, the machine will continue evaluating the polynomial for successive values of x until the memory is filled.

Hence, to terminate this sequence of operations, when the highest desired value of x is reached, the following conditional instruction is required.

R(m). If the number in the accumulator is 0 or positive, continue to the next instruction in sequence. If the number in the accumulator is negative, shift the control to memory location "m" and perform the instruction found there and in succeeding memory locations, accumulator cleared.

Suppose that it is desired to evaluate the above polynomial for values of x up to x = 40, and then break off the sequence.

We will store the number −41 in memory location 30. The instruction in memory location 13 above will not be used and the following will be substituted:

TABLE III.

Memory Location	*Instruction*	*Result of Instruction*
13	A 31	x was stored in memory location 31, hence value of x to be used in next evaluation of polynomial is in accumulator.
14	A 30	-41 is added to value of x to be used next.
15	R 1	If x<(41) say 32, 32 + (-41) = -9, the number in the accumulator is - and the control goes back to memory location 1 and evaluates for x = 32, when we reach x = 41, 41-41 = 0, and the control goes on to instruction 16 to start a new problem.

It can be seen that a relatively small number of instructions can suffice for evaluating a polynomial for x = 2, 3, 4....n. Without facility to modify instructions, n times as many instructions as are needed for one value of x would be required.

The code just illustrated is called a one address code because it refers to one memory location. An example of an operation using a three address code is M (a) (b) (c), which means multiply number in memory position (a) by number in memory

position (b) and store the result in (c). A four address code might be illustrated by M (a) (b) (c) (d), which says multiply the number in (a) by the number in (b), store the result in (c) and go to (d) for the next instruction. The choice among these various ways of designing the control will only be resolved as experience is gained in coding and solution of problems.

II—Applications of Digital Computers.

Any problem can be solved on a large-scale digital computer that can be expressed in terms of the fundamental arithmetic processes and looking up values of functions in tables; moreover, some problems not amenable to analytic solution can be solved on computers by numerical methods. Practical considerations of accuracy required and length of solution time do, however, impose restrictions on problems that can be handled at the present time. As the speed of access time to information and the computing speeds of these machines are increased, the range of problems of interest to the Navy that can be solved becomes larger. On the other hand it is usually wasteful to solve a problem on a large-scale computer that requires less than three or four hours of machine time. Problems which require a large amount of input and output data for a relatively small number of arithmetic operations on these data are in general not done economically on large-scale computers.

Possible military applications of large-scale digital computers include various types of data reduction at laboratories and test stations, solution of the partial differential equations of physics occuring in heat flow and fluid and aero-dynamics, solution of sets of n equations in n unknowns, and use as units in larger systems.

Furthermore, computing machine development is expected to make possible further progress in the purely mathematical theory of non-linear differential equations. It will also make possible a computational approach to quantum mechanical and electrodynamical studies, molecular and statistical studies in chemistry, astrophysical applications, exploratory investigations in theoretical meteorology as well as researches in mathematical statistics and mathematics itself.

Some idea of the times required to carry out an extensive computation on three different types of machines is given in A.M.P. Report 171.2R. The solution of a quasi-linear hyperbolic differential equation is discussed and a comparison is made of solution times on an electromechanical computer, on an electronic machine, the Eniac, and on an acoustic type electronic computer, the Edvac. Such an equation is formulated for describing the pressure on a high-speed, sharply pointed and non-yawing projectile. The knowledge of this pressure enables one to compute the head drag, the air flow between the projectile and the head wave, and of course, the head shape of lowest drag in a given group of head shapes. The mathematical solution of such a problem would eliminate to some extent the experimental techniques carried out in wind tunnels or on firing ranges.

We assume, as given, the shape of the head wave and compute from this the shape of the projectile as well as the flow. The quasi-linear hyperbolic partial differential equations describing the motion of the fluid are:

$$(a^2 - u^2)\, u_y - uv\,(u_y + v_x) + (a^2 - v^2)\, v_y + \frac{a^2 v}{y} = 0,$$

$$v_x - u_y = 0,$$

where (a) is the local velocity of sound, (u), (v), are the components of the fluid velocity at the point (x,y). To make a complete usable table, it will probably be desirable to do about 25 solutions of this problem for different head waves and possibly about 10 solutions for different Mach numbers making the total problem require on the order of 250 solutions of the equations. On the EDVAC the total time of solution will be on the order of eight or nine hours. The ENIAC is able to compute this same problem in 125 hours. An electromechanical machine would take approximately 8,000 hours. It is doubtful whether a single individual using a desk calculator could compute the problem in a lifetime.

The existing large-scale digital computers have made significant contributions in the past. Mark I at Harvard has been used in computing tables of Bessel functions and other functions in the real and complex domains. Employed in solution of systems of simultaneous linear equations, ordinary differential equations,

summing series, it has also solved certain partial differential equations. A difficult study of a heat transfer problem has been completed for the Institute of Mathematics and Mechanics, New York University, under an ONR project.

The Eniac at Aberdeen has been used extensively for calculating trajectories, and has also solved a difficult nuclear physics problem. The new IBM Sequence controlled calculator has been employed in the fluid dynamics problem for the Naval Ordnance Laboratory in connection with work done under an ONR contract.

At a meeting of the Association for Computing Machinery at Aberdeen, Dec. 11 - 12, 1947, Dorrit Hoffleit described the use of the high-speed Aberdeen computers for the reduction of DOVAP (Radio-Doppler Velocity and Position) data taken during V-2 flights at White Sands Proving Ground. In the DOVAP system, a single ground transmitter and three appropriately placed receivers are employed. A transceiver in the missile receives the transmitted signal and retransmits it at double the received frequency. Each receiver compares the signal received from the missile with that received from the transmitter and produces a sine wave record on 35 mm film, the frequency of the wave depending on the missile velocity with reference to the particular receiver. Time markers are provided at 0.01 sec. intervals. The missile coordinates are determined as the intersection of three prolate spheroids having a common focus (the transmitter). The coordinates can be computed at any desired intervals so long as the doppler cycles have been recorded continuously or their discontinuities accurately interpolated. For the first seven trajectories reduced, half second intervals were used, giving 800 points for the longest trajectory.

A comparison of the various facilities employed at Aberdeen for solution of the problem is of interest. Human computers required 15 to 40 minutes per point for the determination of coordinates. This averages 40 or 50 man days for an 800 point trajectory. Although the IBM relay calculator theoretically requires less than a week, actually almost two weeks were required for an 800 point trajectory (including successive differences and velocities). On the Eniac, 1½ to 2 days were required for programming and after that only one second per point was required

for computation of coordinates. It thus required 10 to 15 minutes for 800 points, a time about equal to that of the flight of the missile itself. After it is set up for one trajectory it requires little or no revision for the next. Hence it could do a dozen trajectories in slightly more time than just one. With the new programming recently installed, the set-up time has been greatly reduced.

On the Bell Relay Computer, very little time is required for programming, but 4 or 5 minutes are required to compute the coordinates of each point. This will total about 70 hours for an 800 point trajectory; but, since this machine, unlike the Eniac, can run unattended all night, only 3 working days are required. Thus it is nearly as efficient as the Eniac when missile firings are spaced at long intervals apart and the machines must be set up for each trajectory to be run.

Smoothness of data on trajectories already run, indicates that many fewer points need be calculated particularly after burn-out. In this case, the Bell machine would be superior. On an Edvac type machine, the solution time would probably be an order of magnitude less than on the Eniac.

III—Existing Machines, and those under Development.

The first automatically sequenced computers, the Harvard Mark I and Bell Laboratories machines, are of electromechanical construction. Numerical data and the instructions for operations on these data are introduced through the medium of punched cards or punched paper tape. Computing speeds are of the same order of magnitude, multiplication requiring 2 to 6 seconds and addition 1 or 2 seconds. The great saving of time using these machines is due to the provision for an internal memory and for automatic sequencing which precludes the necessity for a great amount of manual data handling.

The Mark I at the Computation Laboratory of Harvard University was jointly designed by Howard H. Aiken of Harvard and C. D. Lake, F. B. Hamilton and B. M. Durfee of International Business Machines Corporation. It employs many standard business machine parts and is distinguished by great engineering reliability. The main interior memory unit which accumulates as

well as stores has a capacity of seventy-two twenty-three-decimal digit numbers.

A faster and more advanced computer, Mark II, has been completed by Professor Aiken at Harvard under a Bureau of Ordnance contract and has been moved to the Naval Proving Ground, Dahlgren, Virginia. There are twenty-four registers, operated by dial switches, for storing quantities which remain constant throughout a particular computation. The memory unit, which stores 100 ten-decimal digit numbers, is composed of 6-pole mechanically latched relays specially designed at Harvard for the purpose. In order to prevent arcing, the relay circuits are made and broken with cams after the relays have closed or opened. Four relays are required to represent each decimal digit, and are given the values 1 - 2 - 4 - 8, the digit 6, for example, being represented by relays 4 and 2 having closed contacts and 1 and 8 open.

Bell Telephone Laboratories have built four computers employing telephone relays in the memory and arithmetic units. George Stibitz and S. B. Williams were largely responsible for the engineering design. Two of these machines, one at the Naval Research Laboratory and one at Fort Bliss, are employed largely as an aid in the mathematical evaluation of equipment under test. Two larger more general purpose Bell computers were installed at the Ballistics Research Laboratory, Aberdeen Proving Ground and at Langley Field, Virginia (National Advisory Committee for Aeronautics). The larger machine's memory consists of 30 "regular" relay registers for numbers, and 16 registers associated with the arithmetic units, which may be used for storage of numbers under restricted circumstances. The machine employs a "floating decimal point", and a number is then defined to be the algebraic sign, 7 decimal digits, and an exponent (base 10) ranging from -19 to 19 inclusive. A desirable feature is the built-in checking system which causes the machine to stop when a mistake is made and, in many cases, gives an indication as to the cause of the difficulty. The input-output mechanism employs punched paper tape.

Two large scale digital machines employing electronic computer circuits are now in operation: the Eniac and the IBM Sequence Controlled Calculator.

The Eniac (Electronic Numerical Integrator and Computer) was built during the war by the Moore School of Electrical Engineering of the University of Pennsylvania under contract with the War Department, Office of the Chief of Ordnance, and was operated there until moved to Aberdeen in the Fall of 1947.

Twenty ten-decimal digit numbers can be stored in electronic counters (accumulators). Addition time is 200 microseconds and multiplication time is 2800 miscroseconds for 10 digit by 10 digit multiplication. The relatively slow input speed of thirteen and output speed of sixteen ten-decimal digit numbers per second from IBM cards considerably increases the time required to solve most problems.

The Eniac was first designed as a device primarily for solving nonlinear ordinary differential equations. Changes have been made with the intention of increasing the generality of the device so that the Eniac can be used as an all-purpose digital machine. Problems in nuclear physics, air flow, and fluid dynamics have been solved by it. However, it is still efficiently used as a machine for handling fairly simple computations which must be repeated many times, as in the case of calculating firing tables. The generality of the Eniac is limited due to small internal memory, lack of flexibility, and the relatively slow input and output.

The new IBM Selective Sequence Electronic Calculator was first demonstrated to the public 27 January 1948, at the IBM Headquarters in New York. The machine is available for working problems submitted by government agencies as well as by universities and others. It is a general purpose calculator with sufficient memory and flexibility to handle most typical problems. The first problem for which it has been employed was the calculation of the position of the moon, a problem related to one being studied by the Naval Observatory and Yale University in cooperation with the Watson Computing Laboratory under an ONR Mathematics Branch Contract for astronomical research. It has also been employed to solve a shock wave problem for the Naval Ordnance Laboratory and a vibration problem for David Taylor Model Basin.

The memory unit comprises three types of storage: 8 numbers (19 decimal digits plus alegbraic sign) in electronic form, 150

in relays, and 20,000 in card stock tape. There are 66 tape reading stations which can be used interchangeably for numbers or sequencing instructions.

The arithmetic unit is electronic. Multiplication (14 digit x 14 digit to 28 digit product) requires 20 milliseconds, and division (14 digit x 14 digit to a 14 digit quotient) takes 26 milliseconds on the average. Sequencing information is ordinarily introduced through the medium of holes on card stock tape, the instruction unit being a 20-digit, four-address code. One line of coding contains two instruction units; and each line can be introduced to the computer in 40 milliseconds, i.e., 20 milliseconds an order. This time is approximately that of multiplication; hence, the computer is so arranged that waiting for orders does not in general slow up this process. Although the Arithmetic unit adds very quickly, it requires 20 milliseconds on the average to get two numbers from the relay memory, add them together and put the result back in the relay memory because operations can not take place faster than the orders come in. However, one line of coding can be used to order that five numbers be added together in sequence.

One of the important features of electronic computers of the digital type now being designed and built is their flexibility of logical control. Full use is made of the principle that the machine can set itself up at each step of the operation. Problems enter the machine in coded form through the use of paper or magnetic tape, or some other prepared medium. Changes in equations representing particular problems can be readily accommodated as well as changes to entirely different types of scientific problems. After coding a general class of differential equations for example, it would take very little time (one-half to a few hours) to code a particular problem. Moreover, problems and routines of frequent usefulness can be coded on paper tapes or magnetic tapes or wires and stored in libraries for future use. Subroutines that are used in many problems need only be coded once and then inserted at the proper place in each problem.

The electronic computer which seems nearest completion at the present time is the Edvac at the Moore School of Electrical Engineering. It is being constructed under a contract with the

Ordnance Department, U. S. Army, some chassis construction and mechanical work having been done under a subcontract with Reeves Instrument Corporation in New York. The Bureau of Standards provided the input-output equipment. Construction is largely completed and complete system tests are being made.

The acoustic memory consists of 128 eight-word delay lines, the total capacity being 1024 forty-four-binary-digit words. Transfers between the internal acoustic and external magnetized wire memories are at the rate of 29.5 words/sec. Total* addition time is about 800 microseconds and total multiplication time is about 2800 microseconds, on the average. The Edvac is designed with two identical algebraic units, working into a checking system. One algebraic unit may be locked out, in which case the checking is inoperative.

The Department of the Army has set specifications that it must be possible to program the machine to find a single word on the input-output medium, read this word, erase it, and write another word in its place.

The Edvac employs a four address code and has provision for double precision operations.

A computer, the Univac, using an acoustic memory unit, has been designed by the Eckert-Mauchly Computer Corporation, formerly the Electronic Control Company, under a contract with the National Bureau of Standards for statistical use by the Bureau of the Census. Special features have been incorporated to make it suited to the handling of large quantities of statistical data. It is also applicable to the solution of problems of a general nature.

The high-speed acoustic memory has a storage capacity of 1000 twelve-decimal digit numbers.

In order to facilitate sorting and collating, the external memory may employ as many as twelve magnetic tapes, two of which for example, may be used for raw data input, two for output, and eight for temporary storage of intermediate results. Special registers have been provided to enable the shifting of data between low and high speed memories in 60-word blocks (one

*"Total" addition time or "total" multiplication time represents approximately the time required to order two numbers from the high-speed memory, add or multiply them together, and put the result back in the memory.

word = 48 binary = 12 decimal digits) and the transfer of data within the high speed memory in 5 or 20-word blocks. Circuits have also been designed to extract any desired digits from a word.

Transfer rates between the low and high speed memory are expected to be 1000 words per second. Addition is about 350 microseconds and multiplication 1750 microseconds. It includes the time required to obtain the order from the memory and to transfer a number from the memory and add it to or multiply it by a number in the accumulator. A transfer between the memory and the accumulator requires 315 Microseconds.

A contract for the purchase of an electronic digital computer designed for the Office of Naval Research is being negotiated by the National Bureau of Standards with the Raytheon Manufacturing Company. The memory unit of this computer will be of the acoustic mercury-delay type and the input-output units will employ magnetic tape. Performance requirements set for the machine include the handling of such general classes of problems as the solution of non-linear partial differential equations, the problem of systematic sorting, and the preparation of statistical data.

The internal memory as originally planned consisted of 256 17-word mercury tanks, one word in each tank being used for checking purposes. A weighted count of the number of binary ones in a number is made for checking of storage and transfer and is stored with the number. On transfer, the count is separated out, and a new count is made to check that the two counts agree. Arithmetic operations are also checked using an arithmetic weighted count in a process similar to the old method of casting out nines.

It was originally planned that total time for addition, multiplication or division be about 1200 microseconds.

A new design is being prepared for a computer having a 1000-word memory and operating at a 4-megacycles/sec pulse repetition rate. A fixed cycle of operation as employed in the present design will not be used. The computing speeds will be greatly increased.

At the Institute for Advanced Study, Princeton, work goes forward under direction of Professor J. von Neumann on a very high speed, general purpose electronic digital computer embody-

ing the results of previous development in the field. The intended use for this machine is the mathematical investigation of physical phenomena involving, in particular, the solution of non-linear partial differential equations, for which a means of solution has not heretofore been available. This project originally sponsored by the Ordnance Department, United States Army, is being continued under Ordnance contract. ONR, the Air Force and the Atomic Energy Commission are now contributing to the support of the contract.

The anticipated memory unit is the RCA Selectron, or a memory tube adapted from the design of F. C. Williams of Manchester, England, which employs an ordinary cathode ray tube. The Selectron, which is now under development by Jan Rajchman of the RCA Princeton Laboratories, is a vacuum tube containing a non-amplitude-sensitive switching system whereby an electron beam can be directed to a given spot in several microseconds. Within this tube, digits are stored electrostatically in a binary form. A bank of such tubes has the inherent advantage over the acoustic mercury storage that all the digits of a number may be entered or removed simultaneously instead of serially. Moreover, there is no delay in waiting for the proper number to come up in sequence as is the case in most acoustic delay memories.

If the attainment of a satisfactory storage tube requires too long a development period, the first computer model will probably employ a magnetic internal memory with provision for adding storage tubes to this memory when they become available. With a selectron memory, total addition time on the order of 80 microseconds and total multiplication time of 200 microseconds should be attainable.

Considerable research has been done in the field of numerical methods, coding of problems and the logical design of electronic computers. Work at the Institute for Advanced Study on approximation and error theory formerly supported by the Bureau of Ordnance is being carried out under ONR Contract.

A computer is being built at MIT under contract with the Office of Naval Research. A prototype having a memory capacity of 2048 sixteen binary digit numbers, will first be built for testing purposes.

It is expected that the prototype will contain about 3500 vacuum tubes including 32 special electrostatic memory tubes, each storing approximately 1000 binary digits (32 x 32 grid on a 5 in. cathode ray tube). Two banks of 16 storage tubes are to be operated in parallel. To obtain a word from the memory will require two switching operations, one to select the proper bank and the other to position simultaneously the beams of the 16 tubes in that bank to the proper location out of the 1000 possible memory locations.

It is expected that total addition time will be about 40 microseconds and total multiplication time about 50 microseconds.

Sylvania Electric Products is doing some vacuum tube research and circuit construction for the project.

The Air Comptrollers Office, United States Air Force, is supporting the development and construction of a modest scale computer, which has been designed by the National Bureau of Standards. In general, the Edvac design, as modified by Dr. Sam Lubkin, of the Machine Development Laboratory, NBS, has been followed. The mercury acoustic memory will consist of 512 words (64 tanks of 8 words each). The word length is 44 binary digits, with a spacing of 4 pulse times between words. Since the pulse repetition rate is one megocycle per second, maximum access time to the high speed memory is 384 microseconds. To facilitate quick construction, the input-output medium will be teletype tape, but provision will be made for later addition of magnetic tape devices. It is expected that this computer will serve a useful purpose as an experimental device, on which to study engineering design, and that, in addition, it will have considerable power as a general purpose automatic sequenced calculating machine.

The Mark III computer, being designed by Professor Howard Aiken at Harvard under a contract with the Bureau of Ordnance, Navy Department, should be completed by the summer of 1949.

The memory unit will comprise eight magnetic drums of 8-in. diameter on four shafts geared together. These drums are expected to rotate at 7200 rev/min. Each decimal digit is represented in 4 element coded form by presence or absence of a magnetized point in each of 4 channels, a channel being a circle

around the drum. Ten 16-decimal digit numbers together with their algebraic signs are recorded in 4 channels around the drum.

The memory is divided into two parts, one, having a capacity of about 200 numbers, being immediately accessible to the computer unit. This part of the memory is equipped with two magnetic heads per channel, each head capable of reading and writing. The slower part of the memory is about 1600 channels wide (storing 4000 16-decimal digit numbers) and has one magnetic head per channel. By use of a relay selection unit, capable of selecting a set of 4 channels, blocks of ten 16-decimal digit numbers can be read from the low to high speed memory in 20 milliseconds or less.

To simplify preparation of problems for computers, Professor Aiken is developing a "coding machine".

Total addition time will average about 4 milliseconds and total multiplication time about 12.5 milliseconds.

A study has been made by Engineering Research Associates of the effectiveness of several types of recording and reproducing heads for pulse recording on magnetite coated tape bonded to a rotating drum. The heads were displaced from the surface of the tape by 0.001 in. or more to prevent wear. Results have been given for the various heads at different pulse repetition rates and baud lengths. The system performed satisfactorily at a drum surface speed of 1400 in./sec with a linear density of 150 binary digits/in.

ERA is designing a magnetic drum computer and has built many of the units.

Under an ONR Contract, Paul Morton of the University of California, Berkeley, is designing a computer of intermediate scope, which will probably employ a magnetic drum memory. Ease of coding, simplication of circuits and ease of maintenance are to be emphasized.

PART II.

(Reprinted Articles)

The articles contained in this Part II represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the Journal, for articles originally published elsewhere which contribute to the purpose of the Society: to further the advancement of naval engineering.

COMPARISON OF HIGH TEMPERATURE ALLOYS TESTED AS BLADES IN A TYPE B TURBO-SUPERCHARGER.

ACKNOWLEDGMENT.

This article by W. C. STEWART and H. C. ELLINGHAUSEN is a reprint of a paper presented at the annual meeting of the American Society of Mechanical Engineers in December 1948.

ABSTRACT.

The authors describe briefly a series of jet tests, utilizing gas produced from the combustion of diesel fuel oil, as a means for comparing the resistance of a number of high temperature alloys to hot gas impingement. The deficiencies of this method for simulating conditions in a gas turbine are discussed.

The main part of the paper is concerned with comparative tests of a number of high temperature alloys when tested in the form of blades in a Type B turbo-supercharger. The test rotor contained 142 blades representing 12 different alloys. Both wrought and precision cast blades were included. Tests were made at eight temperatures ranging from approximately 1200° to 1500° F., test runs at each temperature being of 50 to 150 hours duration except in the case of the 1500° F. test run which is being continued for 1000 hours.

After several of the test runs, measurements were made to determine the amount of permanent extension in the blades and disc. The extension of the blades and disc accompanying progressively higher testing temperatures is shown graphically. A procedure for correlating the metal temperature of the blades with that of the combustion gas in the nozzle chamber of the supercharger is described.

INTRODUCTION.

During the period 1940-1947 a very large amount of effort and money were expended for the development and improvement of a class of high temperature alloys now sometimes referred to as "Super-alloys". These alloys were developed primarily for

turbo-superchargers, gas turbines and jet engines. For these applications strength and stability at elevated temperatures are considered of first importance. Continued efforts were made to enhance the high temperature strength of these materials. While this was being accomplished, designers were equally diligent in their determination to design for higher operating temperatures and stresses in order to increase efficiency of operation and reduce weight to a minimum.

In this development program, a great many individual alloys were made, and many thousands of tests were conducted on forged, cast and precision cast alloys. Governmental agencies, industrial concerns, educational institutions, and research laboratories cooperated in this work in a most praiseworthy manner. Test data obtained through this cooperative effort are summarized in a large number of reports. Some of these reports were issued by the Office of Scientific Research and Development, some by the Bureau of Ships and Bureau of Aeronautics, and others by private companies. Recently, the classification status for some of these reports was modified, and large portions have appeared as technical publications with the ASTM, ASM and others.

The interest of the Bureau of Ships in these high temperature alloys rests mainly on possible applications for gas turbines. Consequently, test programs sponsored by the Bureau of Ships were directed along these lines. The construction of gas turbines for ship propulsion presents quite different metallurgical and engineering problems from those encountered in the production of superchargers and jet engines. Gas turbines for ships have to be designed for much longer operating life with a minimum of upkeep. Ships machinery must be designed and constructed to resist shock occasioned by gun fire as well as explosions resulting from bombs and depth charges. Parts for large gas turbine units, such as the rotor and casing, are of much heavier section than are similar parts for superchargers and jet engines. These circumstances have introduced special metallurgical and engineering problems in the development program of the Bureau of Ships.

Most of the high temperature test results reported by the cooperating activities were obtained with specimens prepared from small diameter bars, small castings and forgings, and speci-

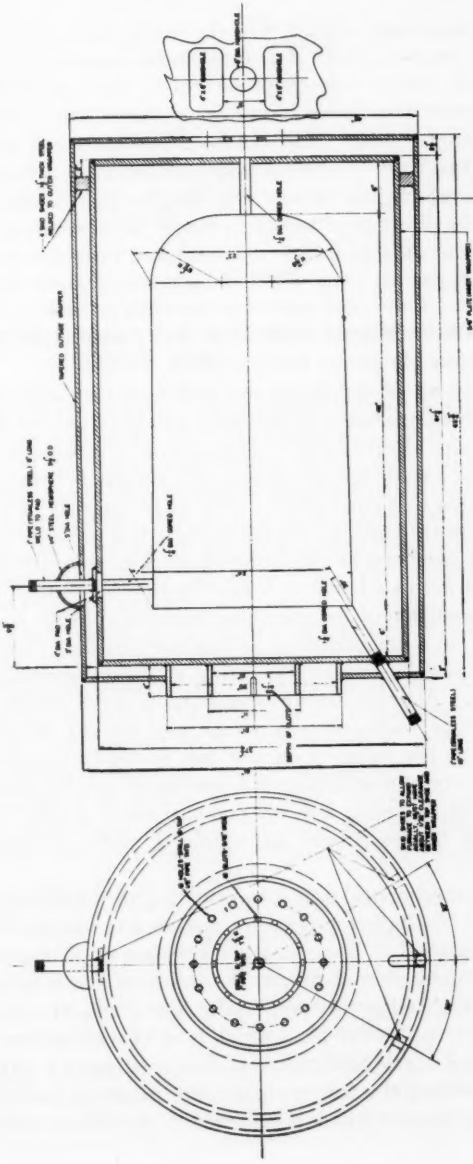


FIGURE 1 - COMBUSTION CHAMBER FOR JET EROSION TEST

mens precision cast to size. On the other hand, an important part of the program of the Bureau of Ships consisted of frabricating relatively heavy forgings of several of the high temperature alloys in order that these might be sectioned and tested, and results compared with those obtained for bar stock and small sections. Likewise, it was necessary to ascertain whether heat treatments, which had proven effective for light sections, were applicable in the case of heavy sections. The welding of large discs presented new probelms that required investigation. The feasibility of casting rotor discs of high temperature alloys was investigated. Later, the work was extended to include the casting of discs with integral blades. At the present time ways for forming hollow blades are being studied.

Practically all of the creep and stress-rupture data reported for the high temperature alloys were obtained in test furnaces containing air. Early in the investigation it was proposed that similar tests be made in a gas atmosphere similar to that produced by combustion in a gas turbine. The practicability of adapting the usual types of creep and stress-rupture test units for conducting tests in combustion gas atmosphere was considered. This examination indicated that it would be difficult to maintain the desired composition of gas not alone simulate velocity conditions encountered in a gas turbine. It was decided that these conditions could best be met by operating a turbosupercharger as a gas turbine. Aircraft turbo-superchargers were readily available, and steps were taken to adapt one for the purpose. It appeared that the value of such a test would be enhanced if a special rotor containing blades of a number of selected alloys was provided. In this way, a direct comparison of several alloys in the form of blades, would be obtained under identical test conditions.

The first experimental unit for simulating the impinging action of high velocity, high temperature gas on gas turbine alloys is shown in Figure 1. The assembly consists of a double wall, cylindrical combustion chamber, the inside of the interior shell being lined with refractory material. The shells are of carbon steel plates except for the head which is of 18:8 chromium-nickel steel. Fifteen nozzle assemblies, each consisting of a nozzle and fixture for holding the test specimen, are mounted radially on a common ring support which is attached to the exhaust end of the

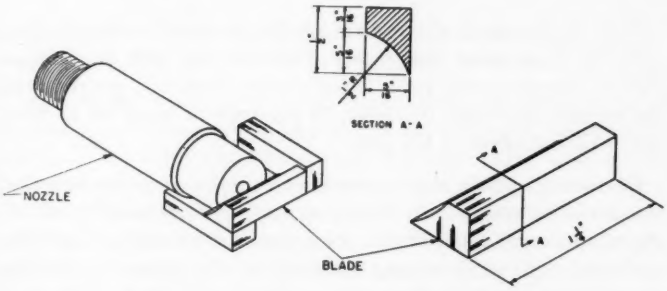


FIGURE 2-NOZZLE AND SPECIMEN ASSEMBLY FOR GAS EROSION TEST

combustion chamber. Details of one of these nozzles and simulated blade section are shown in Figure 2. Gas is produced from diesel fuel oil, combustion taking place under 40 psi pressure. Oil is fed into the combustion chamber from a tank maintained under air pressure. The flow of oil is controlled by means of a pressure operated needle valve actuated from a thermocouple which is mounted in a dummy nozzle. Air is supplied through suitable reducing valves.

Tests were made of a number of alloys both wrought and cast at 1300, 1350, 1400, 1500, 1600 and 1650° F. temperatures. Eight test runs were made at these temperatures, the duration of test ranging from 155 to 1370 hours. The gas velocity at the jets for these tests ranged from 1300 to 1500 ft. per second. Normally, the gas is directed against the apex of the blade so as to sweep upward as shown in Figure 2. However, for one set of specimens the gas was directed at a flat surface of the blade section as well as at the apex. Results of gas analysis for two of the test runs were as follows:

Test Run	Gas Analysis—per cent							
	CO_2	O_2	N_2	CO	H_2	Methane Group	S (calculated as SO_2)	Number of determinations
1	6.2	12.3	bal.	.3	.4	.15	.051	4
6	6.2	12.6	bal.	..	..	...	....	7

The percentages of constituents in the gas remained quite uniform throughout the tests. For test run #8 the carbon dioxide content of the gas was reduced to about 3½ per cent and the oxygen increased to about 14 per cent in order to increase the oxidizing effect of the gas.

Of some thirty-six alloys tested in this series, only one revealed any serious damage. In that case, test conditions of 1350° F. gas temperature and 1380 ft. per second gas velocity caused the test section to swell up and exfoliate at the edges. A similar phenomenon has been observed in connection with another alloy containing a substantial percentage of molybdenum but under less severe test conditions. In this instance disintegration of the alloy occurred during a sealing test carried out in the muffle of an electrical resistance furnace containing air. It should be mentioned that no similar deterioration of these two alloys has been encountered in test units operated as gas turbines.

During one test run, the temperature control mechanism failed to function with the result that the impinged areas of the test sections attained a temperature of approximately 2000° F. for about 20 minutes. This accidental sojourn at excessively high temperature caused Vee shaped notches to be cut in the blade sections in the path of the impinging jet. All materials were affected similarly with the result that no information was obtained as to the relative resistance of the several alloys to hot gas impingement.

At the beginning of the test it was feared that the test sections might be eroded as a result of small particles becoming dislodged from the ceramic lining of the combustion chamber being carried along by the gas stream. Inspection of the edges of the simulated blade sections in the early tests revealed some evidence of erosion from these particles, but with continued operation, trouble from this cause apparently diminished. Should the jet test have indicated important differences in erosion resistance for the several alloys, the role played by non-metallic particles would have been cause for considerable apprehension. However, it was concluded that in this type of test equipment effort should be made so as to eliminate the usual ceramic lining materials from the combustion chamber.

Tests of Blade Materials In An Aircraft Type Turbo-Supercharger.

Experience with the jet tests emphasized the importance of testing gas turbine alloys in hot gas environment while under stress. The adaptation of available creep and stress-rupture test units for operation in combustion gas atmosphere was considered impracticable. One of the perplexing problems was the chemical composition of a gas for simulating service conditions, and the maintenance of a constant supply of the gas over a long period of time. Even with the solution of this problem, no information as to the effect of high velocity gas on metal under stress would be obtained. For these reasons the proposal to conduct creep and stress-rupture tests in combustion gas was abandoned. However, it appeared feasible to conduct such tests in vacuo and oxygen.

The next step was to set-up a Type B turbo-supercharger in conjunction with a specially constructed combustion chamber for operation as a gas turbine. The supercharger rotor provided for test contained 142 blades representing twelve different alloys. The rotor disc is of forged 16:25:6 nickel-chromium-molydbenum (Timken alloy). The supercharger is mounted on a stand bottom-side up from the way it is installed in an aeroplane. The combustion chamber is of the double-wall type approximately 15 feet long by 24 inches outside diameter, made entirely of molybdenum bearing 18:8 chromium-nickel steel by welding. Air for combustion is supplied from an independent blower, and is introduced into the outer space of the chamber near the forward end. Air on entering the outer space flows to the rear where combustion takes place. The combustion gas passes through the inner chamber and enters the nozzle box of the supercharger. This arrangement permits preheating of the air, and at the same time reduces the metal temperature of the inner wall of the chamber. The unit is fired with diesel fuel oil conforming to Navy Department Specification 7-0-2d of 1 October 1943. For purposes of safety the test unit is installed in a concrete lined pit several feet below ground level.

In operating the supercharger, the oil supply tank is maintained under 10 psi air pressure so that oil enters a gerotor pump at this pressure. Oil leaves the pump at 300 psi pressure but a

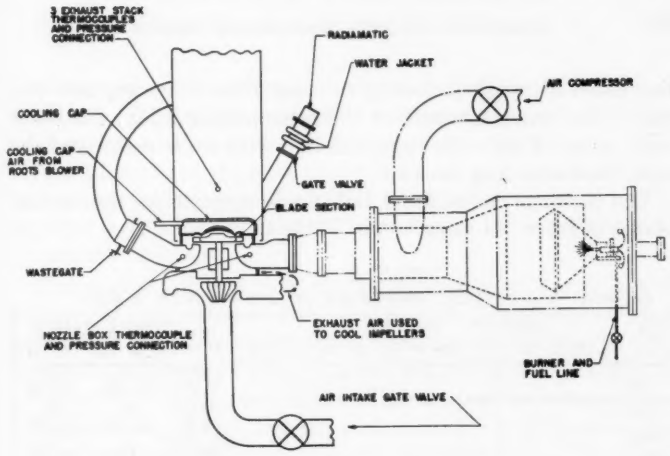
sufficient volume is by-passed to reduce the pressure to the desired value. For 1500° F. operation the oil pressure is maintained at approximately 185 psi. Temperature control is accomplished by regulating a by-pass. The combustion chamber is fired by means of a Babcock and Wilcox Mayflower pressure fuel oil atomizing nozzle and sprayer plate.

Two thermocouples are mounted in the nozzle box of the supercharger. These are placed diametrically opposite, one on the small side and the other on the large side of the box. Thermocouples installed at these locations, at the present time, consist of #14 gage chromelalumel wire contained in porcelain insulators assembled in an 18:8 chromium-nickel steel tube. The hot junction of the element is exposed by means of slots milled in the protective tube. The hot junction bead is encased in platinum sheet metal formed by collapsing a platinum tube around the element. This platinum sheath in conjunction with a chromel wire connecting onto the end of the tube also serves as a support for the element. This thermocouple represents one of the more advanced designs produced by the National Bureau of Standards. Temperature conditions for test are established by means of these two thermocouples.

A third thermocouple, completely enclosed in a 25:20 chromium-nickel steel tube, is located in the gas stream ahead of the supercharger intake flange. This thermocouple serves as a control for operating the unit. Being completely enclosed, it is not subject to conditions which cause deterioration and failure in thermocouples that are exposed to high-temperature, high-velocity gas. The unit is provided with several other thermocouples but these are not required for operation of the equipment. In addition to the thermocouples, a wide angle radiamatic pyrometer is installed in the stack above the supercharger, and focused on the bladed section of the rotor. The length of the major axis of the target area is 1.2 inches covering practically the full width of the blade band.

A schematic diagram of the supercharger installation is shown in Figure 3.

The first specially constructed bladed rotor, designated #1, is shown in Figure 4. Designations A, B, C, and E denote the location of scribed reference circles which serve as base lines for extension measurements. Each blade is numbered for identifica-



TURBOSUPERCHARGER AND COMBUSTION CHAMBER

FIGURE 3

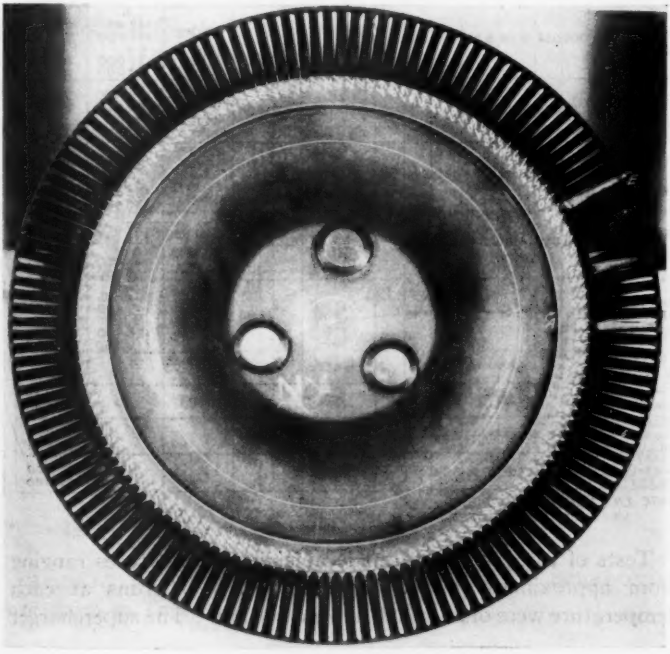


FIGURE 4.
Test Rotor Number 1 Containing Blades of Twelve Alloys.

tion but it is usually necessary to renew these following each test run. The blades are held in the rotor mechanically, the blade roots being of the bulb type, one-half with short necks and the other half with long necks.

The chemical composition and tensile strength for the various blades of rotor #1 are given in Table I.

TABLE I

CHEMICAL AND PHYSICAL PROPERTIES OF SUPERCHARGER BLADES

NAME OF ALLOY	CHEMICAL ANALYSIS												HEAT TREAT	TENSILE STRENGTH OF BUCKETS P.S.I.	
	C	Mn	Si	Cr	Ni	Mo	W	Ti	Cb	Co	N_2	Fe		ROOM TEMP.	1500 °F.
S-495 HT. 22792	0.41	0.53	0.47	13.87	19.89	4.40	3.54		3.51				2300°F 40 MIN. W.Q. 1475 °F. 16 HRS. A.C.	L.N. 152 200 148 950 S.N 132 300 146 500	L.N. 59600 65400 SN 63500 66900
N-155 LOW CARBON HT. 11534	0.15	1.68	0.31	21.50	18.99	2.90	2.08		1.06	17.60	0.06		SOL. TREAT 2240° 1/2 HR. A.C. AGED 50 HRS 1350 °F	L.N. 118 900 119 100 S.N. 115 100 117 900	L.N 66 500 70 300 S.N 64 900 65 100
N-155 HIGH CARBON HT. 11535	0.31	1.58	0.33	21. 12	18.97	2.90	2.17		1.12	17.70	0.07		SAME AS LOW CARBON	L.N. 136 000 134 400 S.N. 134 100 134 000	L.N 75 000 76 500 S.N. 75 650 74 400
19-9DL 810658	0.29	0.89	0.54	19.34	8.51	1.28	2.10	0.22	0.11				50 HRS. 1350° F.	L.N. 132 900 127 600 S.N 127 000 126 000	L.N 54 600 53 250 S.N 56 800 54 000
K42B KB 476	0.05	1.16	0.36	16.78	42.68			2.70		21.23		15.04	2100 °F 1 HR. O.Q. 1500 °F. 20 HRS. A.C.	L.N. 139 400 135 000 S.N. 125 000 127 800	L.N. 98000 90900 S.N. 97400 89 800
S-590 HT. 42883	0.43	0.69	0.09	17.80	19.64	3.82	3.94		2.75	18.37			2300°F 40 MIN. W.Q 1500°F 16 HRS. A.C.	L.N. 129 100 131 000 S.N. 133 900 135 400	L.N. 79 800 75 400 S.N 77 000 71 450
16-25-6 HT. 11542	0.08	1.79	0.58	14.98	25.40	6.00					0.15		24 HRS. 1300°F.	L.N. 137 600 139 250 S.N. 143 000 135 900	L.N. 73 600 70 500 S.N. 65 700 71 300
REFRACT-ALOY HT. M406	0.15	1.21	0.29	20.64	20.71	8.40	4.46			31.41		12.95	2300°F. 4 HRS. O.Q. 1500°F. 240 HRS. A.C.	L.N. 141 000 143 400 S.N. 123 700 115 200	L.N. 97 000 94 500 S.N. 89 400 95 800
CAST 61 ALLOY	0.43			24.66			5.18			67.73			USED AS CAST		72 400 69 600
CAST VITALLIUM	0.25			28.28		5.63				63.84			"		69 000 69 400
CAST 6059 ALLOY	0.39			24.61	33.15	5.34				33.05			"		60 000 63 500
CAST 422-19 ALLOY	0.51			27.56	15.51	6.59				47.61			"		62 000 62 000

NOTE: LN - LONG NECK BUCKET
SN - SHORT NECK BUCKET

Tests of rotor #1 were made at eight temperatures ranging from approximately 1200° to 1500° F. Test runs at each temperature were of 50 to 200 hours duration. The supercharger

was operated at an average wheel speed of 15400 RPM. for all test runs. This speed produces a maximum stress of 10,000 psi at a critical section of the blade due to centrifugal force. The only work performed by the unit other than to overcome friction is that required to drive the impeller. The air furnished by the impeller is directed between the rotor casing and impeller casing to cool the bearings.

The volume of one side of the supercharger nozzle box is greater than the opposite side. Early in the investigation it was found that the gas temperature on opposite sides of the chamber differed somewhat. The average gas temperature was estimated arbitrarily by taking two-thirds of the average gas temperature in the large side of the chamber plus one-third of the average gas temperature in the small side of the chamber. The metal temperature of the blades was estimated to be 15° to 20° F. lower than the average gas temperature. Procedures for determining blade metal temperatures are described later in the paper.

Table II gives a summary of temperature conditions and length of test for the several test runs.

TABLE II
SUMMARY OF TEST CONDITIONS

Test Run	Gas Temperature Conditions—°F.		Duration of Test Run Hours	Estimated Blade Metal Temperature °F.
	Nominal	Average*		
1	1200	1196	100	1181
2	1300	1310	100	1295
3	1350	1347	50	1332
4	1400	1396	50	1381
5	1400	1411	150	1396
6	1450	1464	150	1449
7	1500	1510	100	1495
8	1500	1518	100	1503
9	1500	1514	100	1499
10	1500	1523	146	1508
11	1500	1519	200	1504

* Average gas temperature, see preceding paragraph.

Data for a typical test run (Run 8) at 1500° F. temperature are given in Table III.

TABLE III
TEST DATA FOR RUN #8 AT 1500 °F. TEMPERATURE
DURATION OF RUN 100 HOURS

		Temperature—°F.		
		Maximum	Minimum	Average
Nozzle Box—Small side of chamber		1530	1490	1505
Nozzle Box—Large side of chamber		1540	1500	1525
¾″ from Exhaust Edge of Blades		1460	1420	1430
Temperature 6″ from exhaust side of wheel.	Center of Stack	1325	1300	1315
	½ Radius of Stack	1360	1330	1340
	2″ from Wall	1340	1315	1328
Air Duct Pressure Avg. In/Hg.		10.3		
Stack Pressure Avg. In/Hg.		.4		
Pressure Drop Between Nozzle Box and Stack Avg. In/Hg.		9.1		
Nozzle Box Pressure Avg. In/Hg.		9.5		
Speed of Rotor, RPM		Max. 15,550		
		Min. 15,200		
		Avg. 15,450		
Stress at Critical Section of Blade, psi.		Max. 10,200		
		Min. 9,850		
		Avg. 10,000		

After each test run, the supercharger was removed from the test stand and taken into a constant temperature room where measurements were made to determine permanent extension in the blades and rotor disc. Measurements were obtained on a Zeiss Universal Measuring Machine, and were taken between reference circles E and A, A and B, and B and C. In measuring between circles E and A each blade was measured and the average taken. Two complete sets of measurements were obtained during the course of test run #6. A summary giving results of measurements for the various test runs is contained in Table IV.

The set of measurements designated run #6A was obtained following the first 100 hour test period at 1450° F. temperature, whereas the set designated run #6B was obtained on completion of the 150 hour run. The average extension for each group of blades, obtained by averaging the readings for each blade in the group, are represented graphically on Figure 5.

Table IV
Summary of Blade and Disc Measurements

NO. OF BLADES RUN 8	MATERIAL	TOTAL CHANGE IN LENGTH (INCH)									
		AFTER RUN 2	AFTER RUN 3	AFTER RUN 4	AFTER RUN 5	AFTER RUN 6A	AFTER RUN 6B	AFTER RUN 8	AFTER RUN 9	AFTER RUN 10	AFTER RUN 12
8	VITALLIUM	-.0012	.0000	+.0001	+.0007	+.0010	+.0012	+.0020	+.0025	+.0032	
45	NEW VITALLIUM							+.0024	+.0031	+.0037	
10	ALLOY 61	-.0011	.0000	-.0003	+.0005	+.0005	+.0008	+.0015	+.0019	+.0024	
6	ALLOY 6059	-.0010	+.0001	-.0005	+.0005	+.0005	+.0010	+.0018	+.0022	+.0029	
8	ALLOY 422-19	-.0008	+.0002	-.0003	+.0008	+.0007	+.0013	+.0019	+.0023	+.0027	
9	REFRACTALOY	-.0007	+.0007	+.0003	+.0015	+.0014	+.0016	+.0021	+.0025	+.0029	
8	ALLOY K-42-B	-.0014	.0000	-.0003	+.0003	+.0002	+.0007	+.0012	+.0015	+.0018	
10	HIGH CARBON "N-155"	-.0008	+.0004	+.0002	+.0001	+.0010	+.0013	+.0018	+.0023	+.0027	
8	ALLOY S-590	-.0009	+.0001	+.0001	+.0010	+.0013	+.0013	+.0018	+.0024	+.0028	
8	ALLOY S-495	-.0003	+.0005	+.0004	+.0013	+.0014	+.0017	+.0026	+.0031	+.0037	
10	LOW CARBON "N-155"	-.0007	.0000	-.0001	+.0007	+.0007	+.0011	+.0018	+.0020	+.0026	
12	TIMKEN	-.0005	+.0009	+.0007	+.0021	+.0021	+.0029	+.0039	+.0045	+.0052	
NONE	ALLOY 19-9DL	-.0006	+.0007	+.0007	+.0023	+.0047	+.0080	REMOVED			
	DISTANCE	INCREASE IN/IN.									
	BETWEEN A&B	.0014	.0014	.0013	.0012	.0017	.0012	.0015	.0014	.0014	
	DISTANCE BETWEEN B&C	.0000	.0196	.0136	.0189	.0169	.0166	.0250	.0195	.0210	

On completion of run #6, the 19-9 DL blades were removed from the rotor because of the considerable increase in creep rate following the second period of the run. It was feared that the 19-9 DL blades might fail if tested at higher temperatures with the possibility of damaging the unit.

No new blades of high strength alloys were available for replacement, and a set of Vitallium blades was taken from a standard rotor for this purpose. This proved unfortunate as one of the replacement blades failed during the second 50 hour test run at 1500°F. It was known that the rotor from which the replacement blades were taken had been in service but no other history was available. The failed blade segment damaged a considerable number of blades in the rotor. Forty-five of the damaged blades were removed and replaced with new Vitallium blades. Fortunately, the damaged blades were fairly well dis-

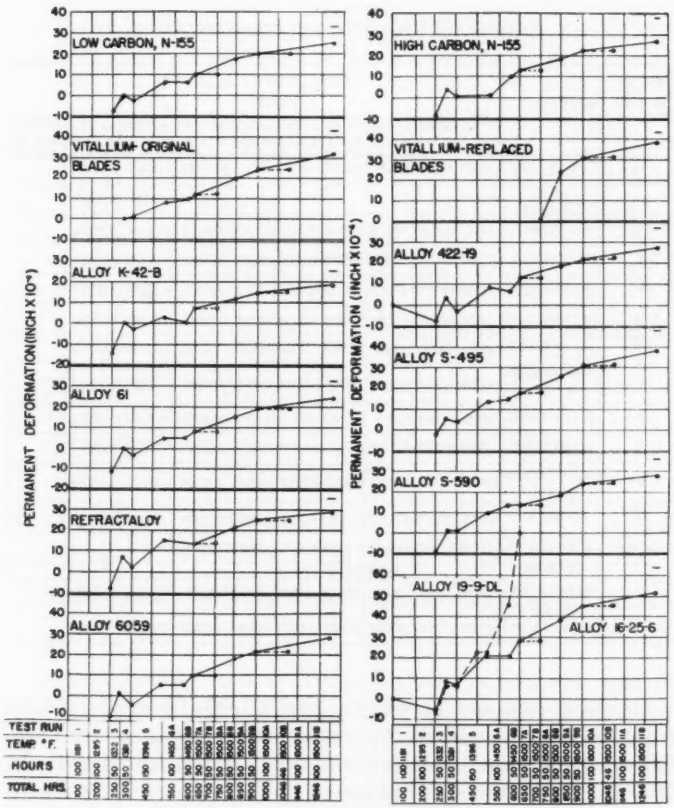


FIGURE 5- GRAPHS SHOWING EXTENSION OF TURBINE BLADES AT VARIOUS TESTING TEMPERATURES

tributed among the several alloys in the wheel so that despite the casualty each alloy was represented by at least six blades.

Renewal of forty-five blades and damage to the reference circles made it necessary to re-establish the reference marks, renew the bearings, and balance the rotor. Then, the test was continued at 1500°F. temperature with the view to obtaining 1000 hours operation at this temperature.

Average extension values obtained for the blades following the several test runs at 1500° F. are also plotted on Figure 5. It is

observed that all of the temperature extension curves show a dip in the vicinity of 1300° F. temperature. This behavior might be attributed to two causes: first, thermal adjustment in the rotor and blade roots may have resulted from the test conditions. Considering that extension measurements were taken between reference circle A on the disc and the tip of the blades, it will be seen that any such readjustment would be reflected in the measurements. Second, precipitation occurs in many of these alloys when held for a considerable period at temperatures in the vicinity of 1300° F., and dimensional changes might have accompanied this phenomenom. However, temperature-expansion curves obtained for a number of alloys of these types have revealed no abrupt changes for relatively short times at temperature.

The short horizontal lines shown on the graphs for each alloy at run #11 represent the maximum extension that would have been obtained had the extension measurements for test runs #7 and #10 not been lost. As previously mentioned, readings for run #7 following a blade failure proved unreliable as did readings for run #10 after a thermocouple tube end failed and rubbed on the blades. Extension rates for these two runs were obtained by taking the average of the rates for the preceding and following test runs.

Gas Temperatures Versus Metal Temperature of Blades.

In order to establish the metal temperature of the blades of the supercharger in terms of the temperature of the gas in the nozzle box, it was necessary to employ an indirect method. The procedure of attaching a thermocouple to one of the blades of the rotor and carrying the leads through a hollow shaft to a set of slip rings was considered. The closely spaced, thin-blade sections are not well adapted for mounting a thermocouple, and it was feared that the blade section might be damaged. However, this method has been employed successfully with larger rotors.

In the following procedure, powders and silver-brazing alloys of known melting characteristics were employed for indicating metal temperatures of the blades. Four blades, located 90° apart were removed from a standard rotor of a Type B supercharger. The concave side of these blades was filled in with

25:20 chromium-nickel weld metal in order to thicken up the section and provide body for an axial hole .136″ diameter. The blades were finished to approximately equal weights and reinstalled in the rotor. Capsules of 25:20 chromium-nickel alloy of the type shown in Figure 6 were prepared for each of the

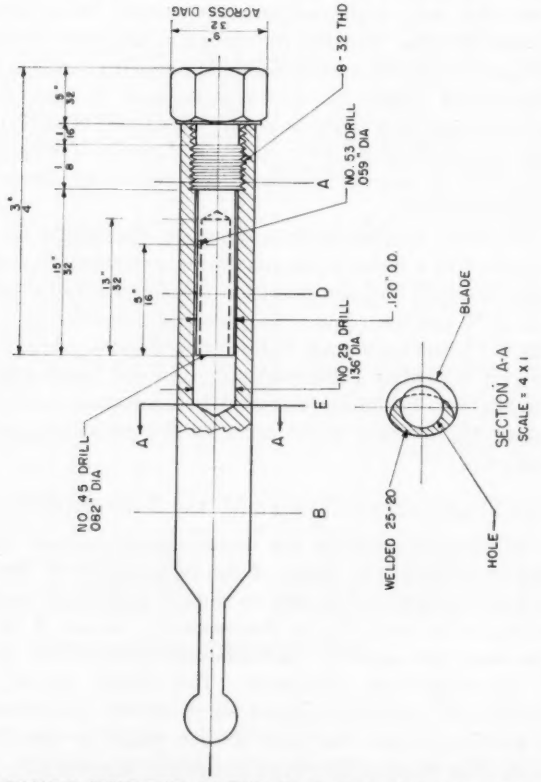


FIGURE 6 - CAPSULE MOUNTED IN END OF TURBINE BLADE FOR HOLDING TEMPERATURE INDICATING POWDER OR ALLOY

modified blades. One end of the tubular capsule is threaded for screwing into the axial hole in the end of the blade. The space within the capsule is for holding either temperature indicating

powders or alloys of known melting characteristics. Centrifugal force tends to force the powder or alloy toward the closed end of the capsule. As long as the temperature indicating material remains in the solid state there is no tendency for it to pass through the small radial hole located near the closed end of the capsule. However, if the material is in the liquid or viscous state it will be forced through the radial hole. Special precautions are necessary to prevent seizure between the threads of the capsule and blade section. This difficulty was obviated by preoxidizing the threads of the capsule before assembly.

Prior to conducting tests in the supercharger rotor, calibration tests were made by placing temperature indicating powders and alloys in similar capsules mounted in steel blocks. These assemblies were heated in a muffle furnace at temperatures ranging from 1300° to 1450° F., and the behavior of the materials observed.

Tests were made at 1336°, 1386° and 1420° F. temperatures in the supercharger. In testing at each of these temperatures, a temperature indicating powder of a different melting point was placed in the capsule of each of the four special blades. The testing temperatures are the average gas temperatures as described previously in the paper. Test results obtained with temperature indicating powders are given in Table V.

Temperature calibration tests also were made of three silver-brazing alloys, the alloys being placed in tubular capsules contained in steel blocks as previously described for the temperature indicating powders. Flow points and melting ranges for the three alloys follow:

Alloy Designation	Flow Point	Melting Range
	°F.	
1	1272	1191-1272
2	1320	1280-1323
3	1345	1339-1345

Three test runs of the supercharger were made with each of these alloys contained in a blade. The fourth blade contained powder prepared from a 1300° F. temperature pellet. The

TABLE V.

Blade Position in Supercharger Rotor	Temperature of Test—Gas Temperature of Supercharger—°F.					
	1336		1386		1420	
	Designated Pellet Temperature °F.	Condition of Pellet Powder After Run	Designated Pellet Temperature °F.	Condition of Pellet Powder After Run	Designated Pellet Temperature °F.	Condition of Pellet Powder After Run
1	1250	(d)	1300	(d)	1300	(d)
2	1300	(d)	1350	(c)	1350	(d)
3	1350	(b) 1/32″	1400	(a)	1400	(c)
4	1400	(a)	1450	(a)	1450	(b) 1/32″

(a) Powder apparently unaffected.
(b) Powder shrank the amount indicated but otherwise solid.
(c) Top and side crust remaining.
(d) All powder had disappeared from fixture.

operating conditions of the supercharger were adjusted to give a temperature of 1320° F. for the test run. Results obtained for the three alloys and the powder after one hour operation of the supercharger are given in Table VI.

TABLE VI.

Test Run (1320 °F.)	Temperature Indicating Materials			
	Silver Brazing Alloys			Pellet
	1	2	3	1300 °F. Rating
1	Alloy melted and ran from radial hole.	Alloy shrank but did not run out.	Intact	Shrank slightly
2	Alloy melted and ran from radial hole.	Alloy melted and ran from radial hole.	Alloy shrank slightly.	Shrank 1/16".
3	Alloy melted down but did not run out.	Alloy partly melted.	Intact.	Marked shrinkage only crust remaining.

SUMMARY.

The need for test information concerning the high temperature strength characteristics and stability of gas turbine alloys beyond that provided by stress-rupture, creep, and gas erosion tests is discussed. The practicability of testing a number of alloys in the form of blades in an aircraft turbo-supercharger operated as a gas turbine is pointed out. By this procedure blades of different alloys are subjected simultaneously to the combined effect of stress and erosion by hot combustion gases after the manner of a gas turbine. This combination of test conditions is not readily attained with the usual laboratory high-temperature testing facilities. An indirect method for evaluating blade metal temperatures corresponding to gas temperatures in the nozzle chamber is described.

Future experimentation is to include quick starting tests of a supercharger, and the testing of a second rotor. The latter contains blades of fourteen alloys, including seven cast and seven wrought materials. Most of these represent alloys of later development than those of the first rotor.

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HIGH PRESSURE, HIGH TEMPERATURE STEAM TURBINES.

Following a discussion of steam flow and gland sealing in contemporary design of large steam turbines, recommendations are made regarding efficiency and operational requirements for construction of reliable medium size and small steam turbines using high pressure, high temperature steam. A new design is described giving test results.

ACKNOWLEDGEMENT.

This article is translated from "Die Wärme" for July 5, 1941. It is substantially complete. This and the following article were written by Dr. Karl Roder and discuss the design and performance of a turbine which shows better performance than that of conventional machines.

The type of turbine described is adapted for use in small and medium sized single cylinder machines as well as in the high pressure cylinder of compound machines. It offers possibility of utilizing the improved cycle efficiency inherent in high pressure and high temperature without the reduction in turbine efficiency encountered in conventional turbine designs for these steam conditions.

With this type of design a series of turbines of different powers may be built, using the same frame, provided the RPM and steam conditions are identical. It is necessary only to vary the blade heights. The Parsons coefficient will be approximately the same for all of the turbines of the series. (Ed.)

In order to obtain perspective regarding the present position of steam turbine design it is first necessary to consider the problems inherent in designing for high pressure and high temperature, and the solutions for these problems. Use of high pressure and high temperature requires that means be provided for taking care of large pressure and temperature differences. In addition, high temperatures necessitate the careful selection of materials.

The small (from a thermodynamic standpoint) specific volume of high pressure steam makes it difficult to design for advantageous flow conditions and creates steam sealing problems. The

specific volume of the steam alone is not a decisive influence but rather the total volume of steam flow ($G \times u$)—the product of weight and specific volume.

The total volume ($G \times u$) determines the size of the flow cross-section and the effect, on efficiency, of the area of the flow passages. These considerations are closely associated with another important problem in the design of steam turbines—the design of efficient small turbines.

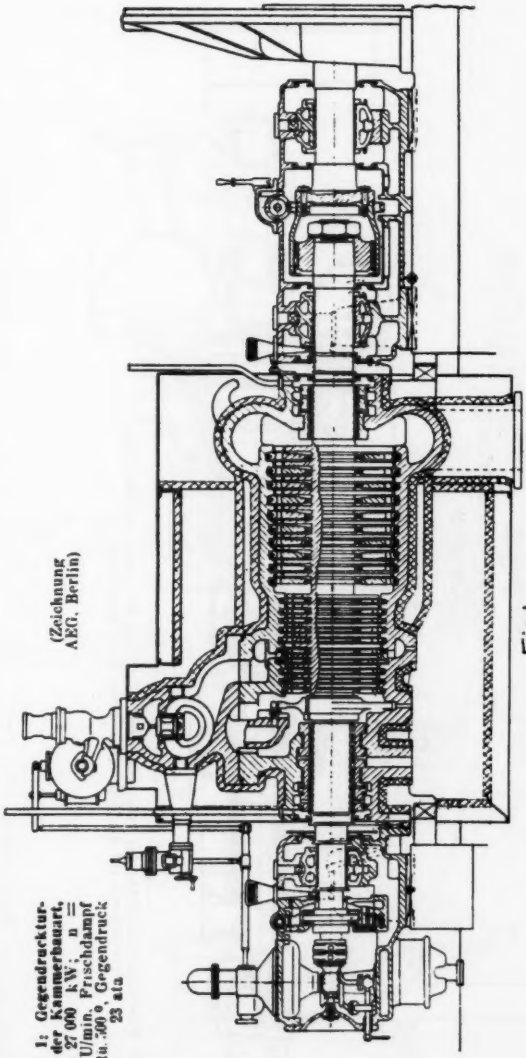
The design problem may be divided into two parts, the first dealing with large steam turbines, the second dealing with small steam turbines.

Large High Pressure Turbines.

Large quantities of steam flow through condensing machines designed for high power, and through back pressure and auxiliary turbines of high and medium power. In spite of the small specific volume of high pressure steam, the flow passages and steam sealing parts for handling these large quantities of steam can be designed in accordance with the principles which for years have led to efficient utilization of small quantities of steam at the large specific volumes associated with conventional moderate pressures. The total volume is about the same in each case.

(*a*) Careful sealing of the interstage pressure drop, accomplished at the smallest possible diameter, is characteristic of the axial, impulse or chamber, type design (Fig. 1) in which the disc-shaped blade wheels revolve in chambers between diaphragms made up of half discs. The diameters at which sealing is accomplished are independent of the mean blade diameter. The steam flow in this design is unfavorably effected by the configuration of the flow path.

After leaving the channels through the blades the flow is no longer guided but a portion of the steam spills over into the space between disc and diaphragm where it flows tangential to the surface of the disc or diaphragm. In this region of unguided flow the jet emerging from the blades draws steam from surrounding space and mixes with it. Packing is required only at the diaphragm between stages and at the high pressure end of the machine.



(Zeichnung AEG, Berlin)

Abb. 1: Gegendruckturbine der Kammerbauart, N = 27 000 kW; n = 3000 U/min. Frischdampf 100 ata, 500°, Gegendruck 23 ata

Fig. 1.
Back Pressure Turbine – "Chamber" Type
27000 KW; 3000 R.P.M.
100 ata, 500°C – 23 ata

Fig.2
Drum Type Condensing
Turbine

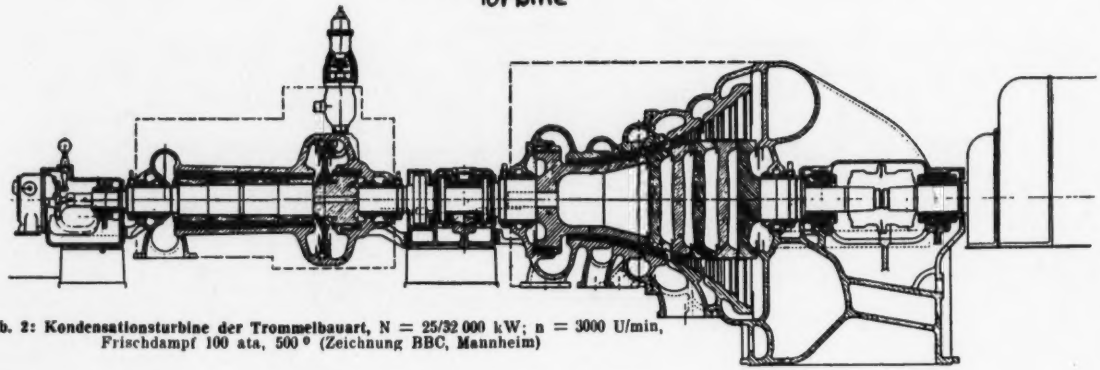


Abb. 2: Kondensationsturbine der Trommelbauart, N = 25/32 000 kW; n = 3000 U/min, Frischdampf 100 ata, 500 ° (Zeichnung BBC, Mannheim)

Abb. 3: Gegendruckturbine der radialen Gegenlauf- bauart, N = 21 000 kW, n = 3000 U/min, Frisch

Abb. 3: Gegendruckturbine der radialen Gegenlaufbauart, N = 21 000 kW, n = 3000 U/min, Frisch dampf 111 ata, 450 °, Gegendruck 14 ata (Zeichnung MAN, Nurnberg)

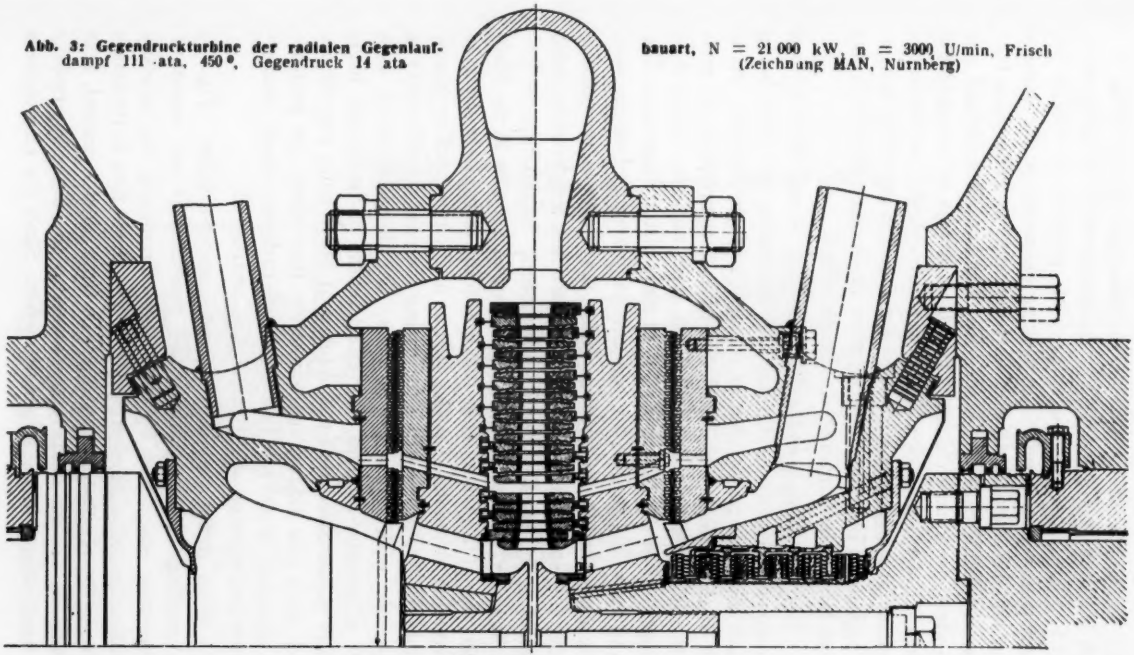


Fig. 3
Radial Counter Rotation Back Press Turbine

(*b*) In the case of the reaction or drum type turbine (Fig. 2), which is usually equipped with a regulating wheel (impulse stage —Ed.), there exists a similar packing problem in which the dummy piston is involved. The packing diameters on the dummy piston and between stages are, however, not, as in impluse turbines, independent of the blade ring diameters but instead are approximately equal to them. Thus disadvantage is compensated by the continuous guidance, along the entire cylinder, of the working steam jet which is an advantage not found in any other design. The regions of unguided flow between blade channels, which are present in all other turbines, are thus avoided.

(*c*) Counter-rotating radial turbines (Fig. 3) are characterized by compact assembly of turbine and generator, the latter being divided into two parts, one on each side of the turbine. As a result of the diameters at which sealing is accomplished the sealing is analogous to that in a drum type turbine; where it is accomplished at the blade ring and dummy piston diameters. The seals in the radial turbine are concentrically arranged on the disc.

The use of a regulating stage inside the casing is not possible and the live steam, acting at full pressure and temperature, must be sealed at the smallest possible diameters. In this type of turbine, however, the live steam must be sealed not only at the blading but also at the faces of the two unloading discs (between fixed and rotating elements—Ed.) as well as at the two packing glands —a total of five places—thus increasing the packing difficulties in comparison with axial flow turbines. On the other hand, as a result of counter-rotation, the effectiveness of individual blade rings is so increased that the number of blade rows may be reduced, each row taking approximately four times the pressure drop absorbed by one row in a drum type turbine without counter-rotation and running at the same peripheral speed.

(*d*) With radial turbines without counter-rotation (Fig. 4) close-coupling between turbine and generator is not necessary. A regulating stage may be used, thus eliminating the necessity of sealing the full admission pressure. The steam flow in the stages following the regulating stage must be from outside inward, thus locating the sealing diameters exposed to the highest pressures and temperatures at the larger diameters. Since the effectiveness

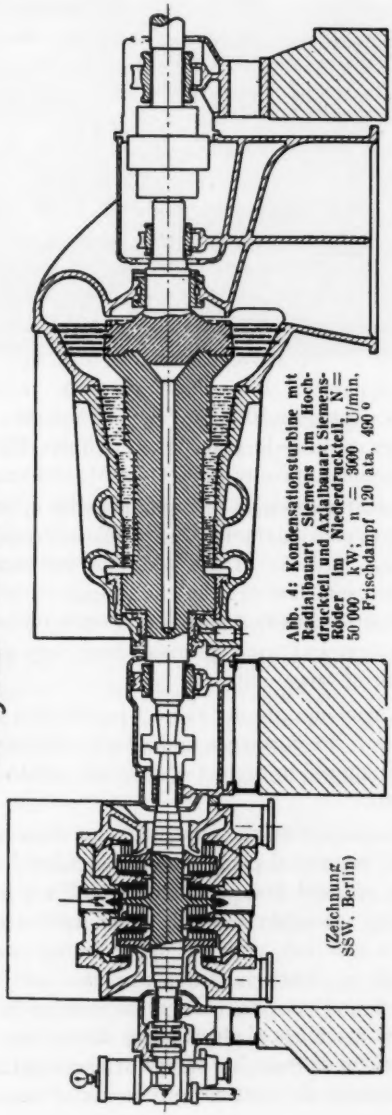


Fig. 4
Condensing Turbine
H.P. Cylinder – Radial Flow, without counter rotation
L.P. Cylinder – Axial Flow

Abb. 4: Kondensationsturbine mit Radialbauart Siemens im Hochdruckteil und Axialbauart Siemens-Räder im Niederdruckteil, N = 50 000 kW; n = 3000 U/min. Frischdampf 120 ata, 490 °

(Zeichnung SSW, Berlin)

of individual blade rows is limited because of the low peripheral speed the machine is designed with a larger number of stages and a corresponding number of discs and smaller pressure drops per stage. In machines designed for very large steam flow the high pressure packing gland and the unloading disc—the most critical part of the turbine—may be eliminated by doubling the number of stages as is done in conventional divided flow low pressure turbines. This doubles the number of blade ring seals. In radial turbines, as in impulse turbines, the region of guided flow is limited to blade channels. The detrimental effect of regions of unguided flow is, however, less with the disc—shaped jets of radial flow than with the annular-shaped jets of axial flow.

In spite of the above differences in design of flow paths and sealing apparatus, large high pressure radial machines have efficiencies between 70% and 80% which are comparable to the efficiencies common in axial flow machines. Radial flow machines may be coupled directly to 3000 RPM. alternators.

Peripheral speed may be combined with optimum pressure drop per stage to provide the Parsons coefficient required for good stage efficiencies. Because of the width of the steam jets the bucket lengths are sufficient to keep end leakage within reasonable limits. Losses due to unsealed glands and exhaust losses are low while rotor friction and windage losses have very small influence upon the output of large units.

These machines are not subject to criticism from the standpoint of efficiency. Performance under load, reliability in operation, and the time required to correct operational troubles are primary considerations.

In this connection the advantages of dividing the casing along the axial horizontal plane should be pointed out (Fig. 5). The internals of axial flow machines, which are divided in this way, are readily accessible once the upper half of the casing has been lifted. In addition, with the upper casing removed but with the rotor still in place, measurements can easily be made between stationary and moving parts thus making it possible in partly assembled machines to check those dimensions which are of great importance in determining safety of operation as well as efficiency. This characteristic of steam turbines is of importance to operators

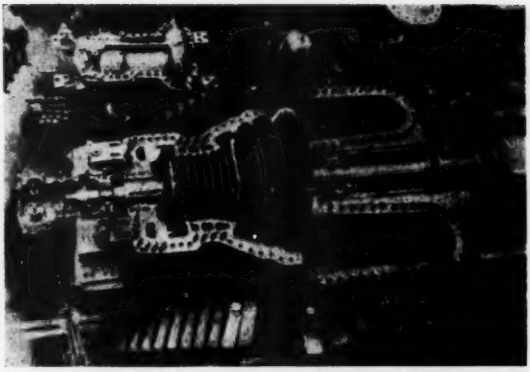


Fig. 5
Axial Flow Type (cover removed)
With Rotor (AEG, Berlin)

since it enables them, with facility and a minimum of time, to check on the operation of a machine, to overhaul, clean, and, above all, to accurately reassemble the machine.

High Pressure Turbines of Medium and Small Size.

When considering the design of efficient turbines for moderate or small powers, involving the design of high pressure stages delivering medium or small output with small or very small steam flow, the following requirements are encountered:

(*a*) Efficiency must be comparable to that of turbines handling large quantities of steam.

(*b*) Construction cost must be proportional to power output.

(*c*) Strength and reliability are of utmost importance since these machines must at all times be ready for immediate operation in spite of infrequent inspection. They must also be suitable for vehicular application.

Referring to (*a*): it is well known that, as a first approximation, the efficiency of a turbine depends upon the Parsons coefficient, i.e. the sum of the squares of the peripheral speeds of all the stages divided by the adiabatic heat drop of the turbine. To obtain a good efficiency it is necessary to have a Parsons coefficient of a

certain value which is independent of the turbine output—a requirement that has no parallel in reciprocating engine design. The choice of proper rotating speed is therefore of considerably greater importance in turbines than in reciprocating engines as is shown in the discussion which follows.

In order to design several turbines for different power outputs but the same rotating speed, which is determined by a direct connected driven machine, it is necessary to use identical stage diameters to maintain the required Parsons coefficient for identical pressure drops and an identical number of stages.

The number of stages and diameter of blade wheels which determine the cost of construction will, in this type of design, be about the same for machines designed for small power output as for machines designed for large output. A reduction in rotor diameter, with rotating speed unchanged, must be compensated by a considerable increase in the number of stages because, for constant Parsons coefficient, the heat drop varies in proportion to the square of the diameter. The essential difference between machines of large and small output is in the length of the blades which has only limited effect upon manufacturing and production costs. (Fig. 6)

The production costs, however, are proportional to the rated power of a machine. For this reason turbines of small capacity must have rotors of small diameter, the peripheral speed being determined, for a given number of stages, by the Parsons coefficient; i.e. the rotating speed must be appropriate to the designed power output and steam conditions. Since the driven machine can seldom be built for high speeds gears are usually used and are considered an essential part of the turbine installation. The cost of the gears must represent only a small portion of the total cost and their reliability must be proven.

Concerning (*b*): it is important that the length of blades in turbines of small power output does not become too small. An unfavorable ratio between contact surface and cross section of flow path, resulting from excessively short blades, would reduce efficiency. Geared turbines alone can be satisfactorily designed and operated at small output and high efficiency. This applies not only to all types of steam turbines but also to all types of turbo-

Fig. 6
Longitudinal Section Thru 3,000 KW and 30,000 KW Steam Turbine

Fig. 6
Longitudinal Section Thru 3,000 KW and 30,000 KW Steam Turbine
RPM = 3,000

Abb. 6: Längsschnitt durch eine 3000-kW- und eine 30 000-kW-Dampfturbine
n = 3000 U/min im gleichen Maßstab

machinery. The reduction ratio, within the limits discussed, is of minor importance in determining the cost of a machine while high rotating speed in a turbine leads to small size and weight, resulting in reduced manufacturing costs, and to larger flow paths and smaller clearances resulting in increased efficiency. Thus the highest possible turbine speed results in the least manufacturing and operating cost.

Concerning (*c*): it is now appropriate to discuss the types of turbine design best suited to high speeds and which type offers the best assurance of reliability.

Adaptability of Known Types of Steam Turbines for High Speed.

The conventional Ljungstrom radial counter-rotating turbine is coupled to the driven machine in such a way that its rotating speed cannot be freely selected. The coupling of its rotors to the shafts of two geared machines is required for a satisfactory application. For this reason it is not a satisfactory high speed turbine.

With the radial turbine without counter-rotation this close association of speeds of driving and driven machines is not necessary. This machine has the disadvantage common to all radial turbines, that discs of small diameter, which must rotate at high speed, offer little space for peripheral blades. Increased RPM will, therefore, result in decreased pressure drop per stage and, accordingly will require an increased number of stages which is undesirable from any point of view.

The axial type, therefore, continues to be the favored high speed machine. The impulse machine, designed for small steam flow, was built long ago by the "Erste Brunner Machine Co." With the acquisition of German patents this concern gradually advanced to the construction of high speed machines. Manufacturers of drum type turbines also resorted to the use of im-impulse stages in turbines designed for small steam flow. These efforts resulted in the decision that high speed machines are deficient as regards efficiency, i.e. steam turbines are best adapted to low rotating speeds. This rule was accepted for steam turbines in general although derived from experience with impulse type

turbines. In my opinion this is a result of the characteristics inherent, in this type of machine, in the construction of the sealing apparatus and flow passages.

The decisive factor in developing a satisfactory impulse turbine is the design of the sealing apparatus. Packing, on small diameters can be designed independent of the blade ring diameter and can be of the necessary configuration.

This advantage decreases rapidly with increase of RPM because, while the diameter of the blade ring can be appreciably reduced, the packing diameter can be reduced only slightly. These two diameters approach each other as the speed increases. As the diameter of blade ring is diminished the flow disturbances in the region of unguided flow, where the steam moves tangential to the rotating discs rather than axially, are increased.

The Drum Type as a High Speed Turbine.

This dependence of flow and sealing processes on diameter as well as RPM does not exist to an appreciable degree in drum type turbines. In this type the diameter at which sealing is accomplished and the blade ring diameters coincide and consequently vary together. The steam does not enter a region of unguided flow after emerging from moving blades. The flow moves between two concentric cylindrical or conical surfaces, proceeding without interruption. The ring-shaped steam jet formed in the first row of fixed blades will persist through the last row of moving blades. In effect the blade rows comb the ring-shaped jet by alternating fixed and moving profile screens. The efficiency of these reaction stages is consequently little dependent upon the blade ring diameter, and, because of the excellent flow configuration, is particularly high. The realization of these flow conditions in high speed machines would necessarily result in performance comparable to that of conventional machines designed for large output.

What of the packing in drum type turbines, where sealing must be accomplished at the relatively large diameters of blade rings and dummy piston and where the inter-stage packing is fitted at the ends of the blades, thus increasing the danger of rubbing which must be carefully avoided?

So far it has been considered satisfactory to leave a radial gap of 1/1000 of the diameter between beveled blade tips and the corresponding rotor or casing surface. The use of blade shrouding, characteristic of impulse blading, cannot appreciably improve the sealing; it leads instead to disturbance of the flow path.

Those parts of the rotor which form the inner boundary of the clearance space are surfaces of revolution which retain their shape during operation. The casing, or outer boundary of the clearance space, consists, however, of upper and lower halves, which, for the purpose of obtaining the necessary tightness, are provided with reinforcement in the form of horizontal flanges at the joint. These flanges are held together by uniformly spaced stud bolts and, upon warming up, the casings are distorted out of round so that internal surfaces of revolution lose their circular form and become oval, elliptical, or egg-shaped. When the casing serves as outer boundary of the clearance space no reliable information can be obtained regarding the phenomena taking place in the clearance space during operation since this space is no longer ring-shaped but has a crescent cross-section of unknown dimensions. Thus, during a given condition of operation the clearances in the horizontal plane of symmetry will differ from those in the vertical plane and each may differ from the manufacturers dimension as checked during assembly. (Fig. 7) Furthermore the casing flanges influence the position and shape of the axis of the casing to the extent that the axis of the casing and the rotor do not coincide once the machine has been warmed up. Uncertainty regarding the varying configuration of the clearance space leads to uncertainty regarding clearance dimensions and results in considerable uncertainty regarding performance in operation.

These considerations show that this type of casing is not suited for use as the outer boundary of the clearance space and so its use as such must be abandoned. It was possible to employ this type of casing during the last decade only because it was associated with moderate steam conditions and relatively large clearance spaces. Occasional rubbing, particularly during improper and rapid warming up, results in increased clearances leading to a proportional increase in steam consumption during operation.

Abb 7: Längsschnitt durch die größte bisher gebaute 3000 tourige Turbine, N = 60 000 kW; n = 3000 U/min (Bauart Siemens-Röder)

Fig. 7
Section Thru Largest (up to July 1941) 3,000 RPM Turbine.
Rated Load = 60,000 KW. Design, Röder-Siemens.

Solving the Sealing Problem in the New Drum Type Design.

The outer boundary of the clearance space formed by fixed blades and packing demands the use of special parts which must satisfy the following requirements.

1. They must be surfaces of revolution which, like the rotor blading, must remain surfaces of revolution unaffected by casing distortion during operation.

2. Their axis of symmetry must, under all conditions of operation, coincide with the axis of the rotor.

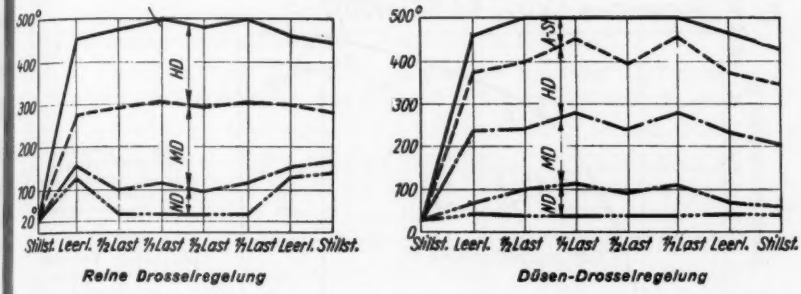
3. Temperature variations in the operating medium (steam) must not increase clearances.

The first requirement is fulfilled by the new drum type turbine because the fixed blade support and packing support is made in two parts, each designed as a surface of revolution and held together by additional parts so that the shape as a surface of revolution remains unaltered. These parts are free to expand when heated and are concentric to the casing so that they fulfill the second requirement of maintaining symmetry about the axis of the rotor. Symmetry of form is obtained by eliminating flanged joints, i.e. by a construction composed of parts disposed symmetrically about the axis. Piping connections are not made to this part of the turbine.

The fulfillment of these requirements and the resultant effect upon turbine performance is shown by the temperature variations within a machine during operation. (Fig. 8) The large increases in temperature taking place during warming up and upon application of load appear in striking contrast to the small temperature variations which occur during normal operation.

Transient temperature differences between the parts which form the inner and outer boundaries of the clearance space occur during the rapid increase of temperature during warming up because the rate of heating varies between different parts. These temperature differences, the magnitude of which depends upon various factors, cause variations in the dimensions of the boundaries of the clearance space which determine the clearance. The temperature differences in the metal parts are small in comparison to the temperature variations in the steam and, as shown in Fig. 8, occur only

Abb. 8: Betriebsmäßige Änderungen der Dampftemperaturen im Hochdruck-, Mittel- und Niederdruckteil einer Hochdruck-Kondensationsturbine mit Drossel- und Düsenregelung (Zeichnung VDI-Verlag*)



*) K. Röder, Die Abdichtungsaufgaben im Dampfturbinenbau, Arch. Wärmewirtsch., 18. Jg., 1937.

Fig. 8
Operational Steam Temperature Variations in the H.P., I.P. and L.P. Cylinders of a High Press. Condensing Turbine, with Throttling and Valve Cut-out Regulations.

while turbines are warming up, i.e. during he first phase of each period of operation.

The requirement mentioned earlier is fulfilled when the fixed blading and packing support forming the outer boundary of the clearance space, expand faster with increasing temperature than does the rotor forming the internal boundary. Thus the variations in clearance which occur during warming up and application of load are such that the danger of rubbing during this critical phase of operation is eliminated. A long warming up period is therefore not required nor are the operational errors which may occur during application of load dangerous. The clearance need only be checked for the greatest temperature differences (in the metal—Ed.), which are small in proportion to the temperature variations (in the steam—Ed.), and result in extremely small changes in the dimensions of the parts which determine the clearances.

Results of the New Design.

Using the conventional clearance of 1/1000 of the diameter in the first 1000 KW. turbine which was built in accordance with the foregoing requirements and tested by the Tech. Hochschule

Fig. 9
Internal Efficiency of Turbine at Technl. College, Hanover, compared with Turbines Tested by K. Jaroschek [VDI, Bd. 82 (1938)]

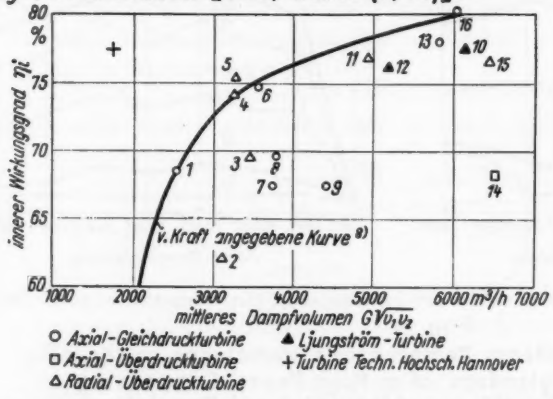
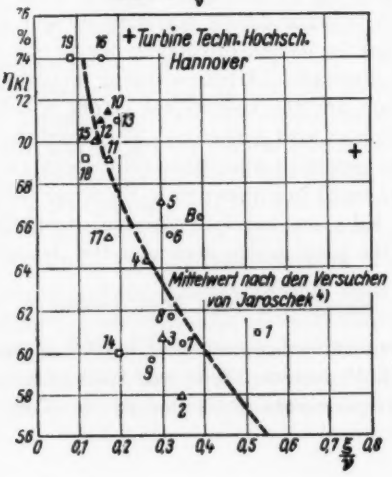


Abb. 10 (rechts): Klemmenwirkungsgrad der Turbine der T. H. Hannover, über der Volumenkennzahl $\frac{\xi}{\nu}$
(Zeichnung Franckhsche Verlagsbuchhandlung)†)

Abb. 9: Innerer Wirkungsgrad der Turbine der T. H. Hannover, verglichen mit denjenigen der Turbine nach den Versuchen von K. Jaroschek, Z. VDI, Bd. 82 (1938), S. 993
(Zeichnung Franckhsche Verlagsbuchhandlung)†)

†) K. Röder, Ein Beitrag zum Bau von Hochdruckdampfturbinen, Elektrizitätswirtschaft, 39. Jg., Nr. 20 vom 15. Juli 1940.

Fig. 10
Terminal Efficiency of the Tech. Col. Hannover Turbine vs. Volumetric Coef. $\frac{\xi}{\nu}$



Hannover with steam at 50 atmospheres and 450° expanded to 1.3 atmospheres, an exceptionally high degree of reliability was obtained. This was substantiated in the high pressure turbine test laboratory of Bruckner, Kanis and Co., in Dresden.

The time required to start the machine and reach the normal speed of 10,000 RPM. was reduced to 5½ seconds and the time required to go from stand still to full load was 25 seconds. Satisfactory operation was obtained regardless of the temperature of the machine at the time it was started, i.e. regardless of the length of time it had been out of operation.

Rapid changes of temperatures as great as 200°C. resulting from sudden shifting from superheated steam (450°C.) to saturated steam and vice-versa, with an intervening period of steady state operation, resulted in no change in performance. The back pressure during steady state operation varied between 0.5 and 3.0 atmospheres.

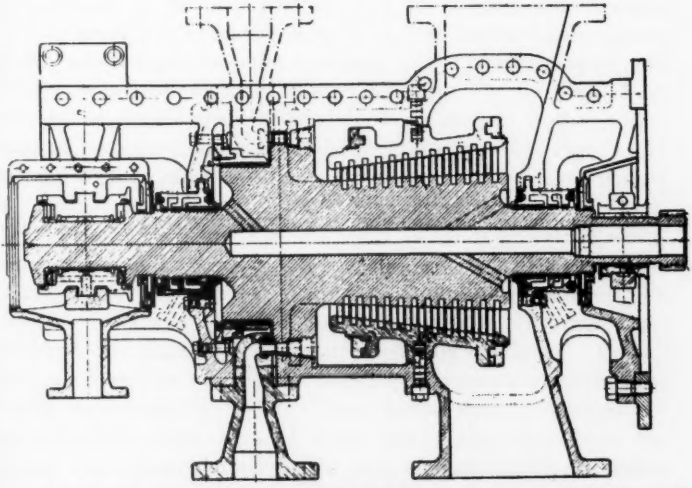


Abb. 11: Längsschnitt durch die Gegendruckturbine der neuen Trommelbauart für hohe Dampfspannungen und hohe Dampftemperaturen

Fig. 11
Section Thru New Drum Type Reaction Turbine for High Pressure and High Temp. Steam

The internal and overall efficiencies which, as shown on Figs. 9 and 10 respectively, are substantially higher than those of other designs, remained unchanged during these performance tests. When the turbine was opened there was no evidence of rubbing nor were any parts so distorted as to be in contact.

The casing and rotor of this 1000 KW. turbine weighed 620 kg, ie 0.62 kg/KW., thus requiring a relatively small amount of steel.

These high speed drum type turbines, with sealing apparatus and fixed blades installed (Fig. 11) in accordance with the methods described above, show not only considerably higher efficiency than any other type built to date but also require a minimum of steel and entail a minimum cost of construction; and, most important, may be put in operation on short notice without special steps to avoid operational failures.

NEW HIGH SPEED HIGH PRESSURE STEAM TURBINE.

ACKNOWLEDGMENT.

Translated from "Verein Deutscher Ingineure"—1939.

The extracts from this paper give further information relative to the design and performance of the new type turbine discussed in the preceding article.—Ed.

NEW DESIGN CRITERIA.

While in axial flow reaction turbines the fixed blades are generally secured to grooves in the casing the new type turbine provides special blade supports as shown by "a" in Figs. 1 - 3. These supports remain circular during operation although they are made up of two halves. Oval distortion of the part supporting the fixed blading and the especially dangerous pinching at casing flanges which leads to rubbing, both common in conventional designs, are eliminated. The conical blade support shells, which form the outer boundary of the clearance space, are held together by specially shaped members, "b" in Figs. 1 - 3. These ring shaped members hold the blade support shell as a surface of revolution and support the shell at plane "c" (Figs. 1 - 3) normal to the casing and free to expand so that deformation of the casing cannot be transmitted to the blading support shell.

The blading support shell responds to the temperature variations of the steam more rapidly than the rotor. The radial clearances, therefore, increase with increasing steam temperature which, in starting, amounts to 400° to 500°C. (Temperature drops of this magnitude do not occur during operation). During normal operation, including securing, temperature differences rarely exceed 100° to 150°C. These temperature differences are not exceeded when slugs of water are carried over from the boiler. The greatest temperature drop encountered in the steam during operation is only ⅓ to ¼ the temperature rise encountered on starting. By moving the outer boundary of the clearance space inward in this ratio smaller clearances are obtained than are found in conventional designs where the outer boundary of the clearance space responds to steam temperature more slowly than the rotor.

Fig. 1-3
Construction of Guide Blade Holding Element of Axial Reaction Turbine

Bild 1 bis 3. Einbau von Leitschaufelträgern in axiale Überdruckturbinen.
a Leitschaufelträger b zum Zusammenbau der Leitschaufelträger dienende Einbauelemente c Zentrierebene im Gehäuse

Reduction of clearances of this magnitude are not necessary to obtain clearances which result in favorable steam consumption in comparison to clearances used to date. Efficiency and reliability are, therefore, simultaneously improved with respect to steam turbines on the market today.

Since the clearance increases when a turbine is started no dangerous condition of operation occurs during the initial loading as is the case in conventional designs where the minimum operating clearance is unknown. In conventional designs the clearance decreases during the application of load because the outer boundary of the clearance space becomes oval. In the new type of design the starting clearances are greater over the entire periphery. There is nothing, therefore, to prevent rapid starting of the turbine.

Since, during steady state operation, the clearances return to their normal designed dimensions after all parts have been warmed up, leakage losses, which have an important effect on efficiency, are small and excellent guidance of the steam is afforded contributing to high efficiency in the reaction elements.

The basic rule "small clearances when cold and during steady state operation, increasing clearances during transient conditions dangerous as regards safety of operation" leads to accurate construction, small leakage losses during operations and minimum danger of internal rubbing.

Small blade and packing clearances require that bearings have small clearances. This may be accomplished if the bearings are protected from large variations of temperature during operation.

The bearing supports, heated by the steam within the turbine, must be adequately cooled by such means as cooling of the hollow shaft or the circulation of oil or air to carry off heat which flows from the casing into the bearing supports.

Small packing diameters are just as important as small blade sealing diameters. They exercise a much greater influence on the efficiency of a high pressure elements of a turbine than on the low pressure section. In condensing machines, therefore, the high pressure section should be separated from the low pressure section, the high pressure turbine being designed for high speed and small size, and connected to the low pressure turbine through gears.

Design and Performance of the Roder Turbine.

The Roder turbine was designed for 1000 KW. at 16,000 RPM. using steam at 46 atmospheres, 430°C., and exhausting at 1.3 atmospheres. It has one impulse and 14 reaction stages. The sum of the squares of the velocities of the wheel speeds is 426,000 $(M/S)^2$. (The total combined stress in the rotor blades is 21,300 psi at full power, of which 4266 psi is bending stress due to steam loading.—Ed.)

The turbine is equipped with throttle control but nozzle control could be provided since the first stage is an impulse stage. There are two groups of nozzles 180° apart. Blades are not shrouded but are tapered at the ends in the conventional manner. The casing is

TABLE I

Test of Impulse-Reaction Turbine (Back Pressure) at the Technische Hochschule, Hannover.

Brake power - KW	193	378	557	789	823	974
R.p.m.	16160	16080	15940	16100	16320	16320
Total Steam - kg./hr.	1917	2947	3955	5109	5230	5947
Gland leakoff - kg/hr.	24.0	27.8	32.7	37.6	38.5	41.0
Vapor leakoff - kg./hr.	6.8	9.0	9.5	10.1	10.4	10.7
Pressure ahead of nozzles, P_1 - ata	15.0	22.9	30.3	39.5	40.8	46.1
Temp. " " " - °C	402	417	417	427	428	434
Wheel Chamber press. - ata	4.0	6.0	8.0	10.5	10.5	12.0
Back pressure, P_2 - ata	1.36	1.36	1.37	1.36	1.36	1.36
Exhaust Temperature - °C	195	161	134	117	115	108
Total adiabatic drop, H_o - $\frac{\text{K cal.}}{\text{Kg.}}$	134.4	154.2	164.4	175.9	177.3	182.7
Adiab. drop of impulse stage - $\frac{\text{Kcal.}}{\text{Kg.}}$	83.3	85.2	84.8	84.4	86.0	85.7
Therm. eff. referred to steam before nozzles & B.KW.---%	64.3	71.5	73.6	75.5	76.4	76.9
Internal Eff. (η_i) of turbine (from steam press. & temp.) - %	69.7	74.3	75.8	76.8	76.9	77.5
Internal Eff. of impulse stage - %	58.2	59.0	58.5	60.1	59.1	59.9
Internal Eff. of reaction stages - %	76.4	82.2	83.2	82.6	83.6	83.8
Steam consumption - kg./H.P.-hr.	7.32	5.74	5.22	4.76	4.67	4.49
Parson's Coef. $\varphi = \sum u^2/H_o$ - $Kgm^2/Kcal\text{-}S^2$	3230	2780	2570	2450	2500	2430
Hydraulic Coef. $\nu = \sqrt{\quad}/91.5$	0.621	0.577	0.554	0.541	0.548	0.539
Mean steam vol., $G\sqrt{v_1 v_2}$ $m^3/hr.$	1107	1332	1490	1660	1670	1770
Volume Coef. $\xi = (p_1 p_2)/GH_o$ Kg.hr/Kcal.m^2	0.530	0.475	0.445	0.425	0.426	0.413
Melan's number ξ/ν Kg.hr/Kcal.m^2	0.853	0.823	0.803	0.786	0.779	0.766

sealed by radial stuffing glands of the type first proposed by Ljungstrom and approved for use in SSW (Siemens-Schuckert) radial turbines.

The tests of the turbine were conducted by K. Jaroschek, V.D.I. The turbine was direct-connected to a 16,000 RPM. hydraulic brake. Test results are shown in Table 1.

Fig. 4
Section Thru H.P. Turbine of Tech. College, Hanover.

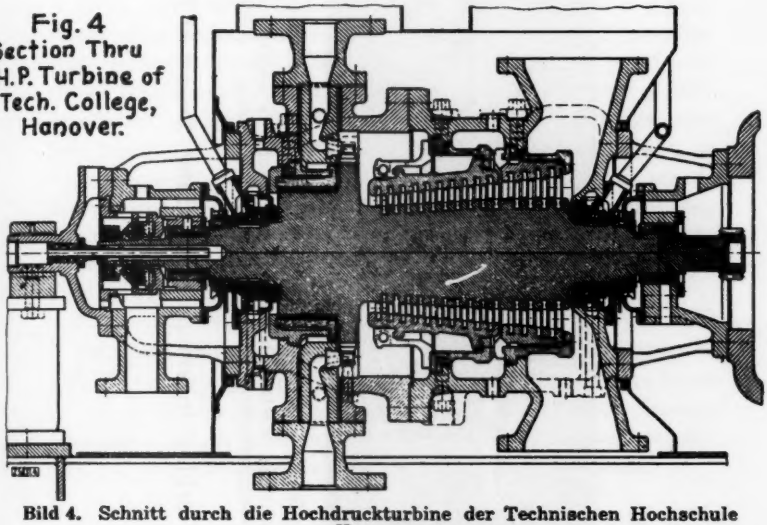


Bild 4. Schnitt durch die Hochdruckturbine der Technischen Hochschule Hannover.
1000 kW, 16 000 U/min, Frischdampf 46 ata und 430°, Gegendruck 1,3 ata.

The efficiency of the turbine changes relatively slowly as load is changed because the machine is driven by varying the inlet pressure. The efficiency of the impulse stage remains nearly constant: that of the reaction stages decreases rapidly only at light loads.

Fig. 6 (Fig. 6 is not reproduced here because it is the same as Fig. 9 of the preceding article) shows the internal efficiency with respect to inlet steam consumption.

For purposes of comparison Jaroschek's test results were plotted together with those of 24 industrial steam turbines as well as the Kraft curve which runs through the region of optimum values.

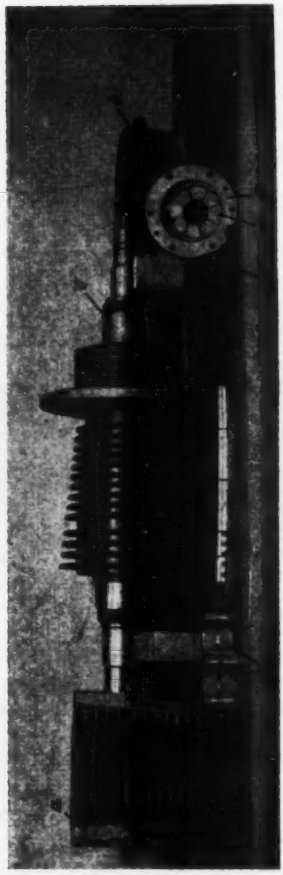


Bild 5. Ansicht der Innenteile der Turbine nach Bild 4.
a ND-Leitschaufelträger *b* HD-Leitschaufelträger *c* Läufer *d* Traglager-Schalen *e* Dichtungsträger des Ausgleichkolbens *f* Laterne des vorderen Lagers *g* Michell-Druckklötze

Fig. 5
View of Turbine Internals of Turbine Shown in Fig. 4

The results show that the efficiency of the new turbine is substantially higher than that of other designs.

Assuming the turbine were connected by a simple reduction gear to a 1500 RPM. generator, with a 2% throttle loss at the main cut-out valve, a 3% gear loss, a 5% generator loss and 1000 KW. output, the thermodynamic efficiency, based on output at the generator terminals, will be about 70%. This value is extra-

ordinarily high in view of the high steam pressure (46 ata.) and the small steam consumption (5.9) t/h shown on Fig. 7.*

Fig. 7 (Fig. 7 is not reproduced here because it is the same as Fig. 10 of the preceding article) shows the terminal efficiency plotted against the ratio E/u suggested by H. Melan, V.D.I. in which $E = (p_1 - p_2)/GH_0$ is the volume coefficient and $u = \sqrt{u^2/H_0}\ /91.5$ is the hydraulic coefficient. This gives a definite relationship between efficiency and Melans number E/u rather than scattered results. The Meininghaus—SSW type turbine shows performance superior to that of the other turbines plotted, its efficiency being higher than the others. The new design reaction turbine is even better, however. Plotting these results in Figs. 6 and 7 shows clearly the efficient conversion of energy accomplished in this turbine as originally designed and using steam of low specific volume. Surprisingly high efficiency is obtained with this turbine which is small, light, and inexpensive to produce.

After extensive brake tests and 500 hours of operation with a generator during which 260,000 KWH. was generated, the turbine was opened and all parts, including those at the boundaries of the clearance space, showed no evidence of damage. One may conclude therefore that small leakage losses and practically constant efficiency were maintained during operation.

STEAM GUNBOAT MACHINERY — A LIGHT-WEIGHT STEAM PLANT.

SUMMARY.

The machinery for the Steam Gunboats was designed and produced in a period of about fifteen months following July, 1940. It was required to be to the standards of a reliable marine steam plant but less than half the weight per SHP. of existing practice. The paper describes the machinery broadly, and indicates generally how the target weight figure for the installation was achieved.

ACKNOWLEDGMENT.

This article is reprinted from a paper presented at the autumn meetings of the Institution of Naval Architects by Comdr. (E) H. A. K. Lay, RN and Comdr. (E) L. Baker, RN of the Engineer-in-Chief's Department of the Admiralty.

The steam gunboats (S.G.B.s) were planned to counter the E-boats operating in the Channel after the fall of France and were to have steel hulls of about 115 ft. length and a total of 8000 SHP. on two shafts. Steam was chosen as petrol engines were not available and as it was considered that less maintenance, better silence, greater reliability and a lower fire risk would be obtained. No compression ignition engine was available with the required power-weight ratio.

In order to obtain the required performance from this vessel a limit of weight was set which many engineers thought would be impossible with steam machinery, particularly as the policy was laid down that design stresses in excess of those known to have given satisfactory results could not in general be exceeded and that owing to the supply position no light weight alloys could be used.

Broadly, the machinery installation consisted of a boiler supplying steam for two shafts, each of which was driven by its own single cylinder turbine through single-reduction gearing. A simplified closed feed system was employed. To help in achieving the necessary low specific weight of the machinery,

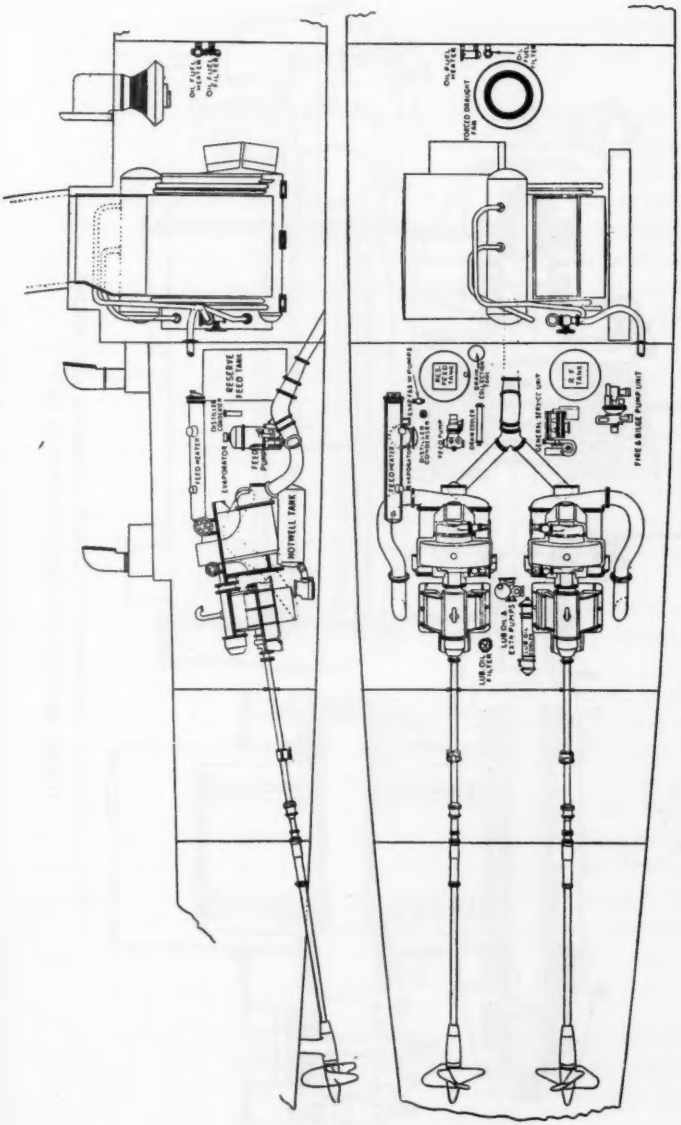


FIG. 1.—GENERAL ARRANGEMENT OF MACHINERY SHOWING FOSTER WHEELER BOILER

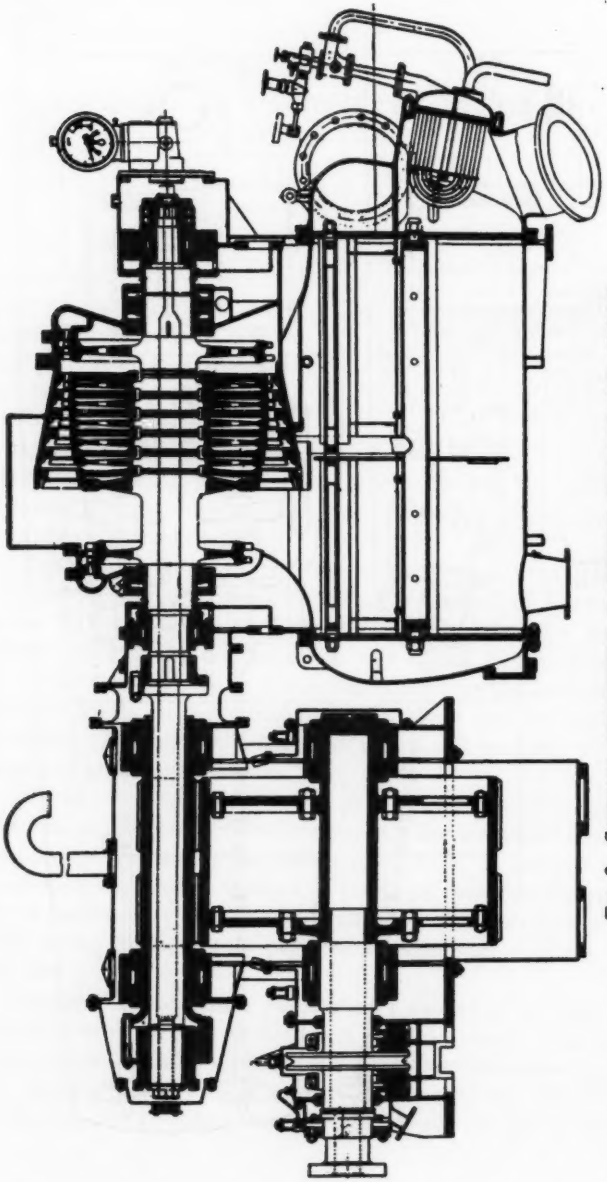


FIG. 2.—SECTIONAL ARRANGEMENT OF TURBINES, CONDENSER AND GEARING (MESSRS. METROPOLITAN-VICKERS ELECTRICAL CO.)

duplication of auxiliaries for standby purposes was avoided and, as a general policy, no emergency arrangements to cover machinery damage or defects were fitted.

A long life was not foreseen for the S.G.B.s, and allowances for wastage of the material by corrosion were reduced or omitted.

The original requirement for these vessels was an endurance of 200 miles at full power. In view of the functions to which it was then intended to limit them, cruising radius and, consequently part-load performance of the machinery, were not of importance. The design of the main turbines, therefore, was free from one of the chief problems which normally vex the designers of warship engines.

It was hoped that the steam gunboats would achieve about 38 to 40 knots, but during the brief period of their design and construction requirements changed; the need for adding radar apparatus and radar personnel, and other equipment, and a desire for greater full power endurance led to an increase of displacement of nearly fifty per cent. The maximum speed was thereby somewhat reduced: but the performance of the propelling machinery was, of course, not in itself affected by the alterations.

The propeller speed—900 RPM. at full power—was satisfactorily high from the point of view of the machinery designer intent on weight reduction.

For various reasons it was decided to use Pool diesel oil as the standard fuel; this decision was slightly modified as a result of service experience, but it did to a very small extent assist in reducing the weight of the machinery.

First thoughts were given to the design in June 1940 and a serious start made about two months later; the first S.G.B. left Messrs. Yarrow's yard on a successful preliminary trial in November 1941. During this brief period of fifteen months the complete and rather novel machinery installation was designed, constructed, and installed.

The Space Available for the Machinery.

The beam of the vessel in way of the machinery spaces varied from about 23 ft. to about 21 ft. at the after-end of the engine-room; but the framing and turn of the bilge, of course, considerably reduced the internal width available.

The height from the ship's bottom to the deck over the machinery spaces was about 11 ft.; while this was increased by casings over both engine- and boiler-rooms it was severely encroached upon by double vertical keels about 4 ft. deep running the length of the compartments; and also by the deep transverse framing. The total length of the machinery space available was about 39 ft., divided by one transverse watertight bulkhead into engine-room and boiler-room.

Weights.

The difficulty of fitting suitable machinery into the space provided was not inconsiderable, but was not so great as that of getting down to the target weight. The really notable feature of the installation, and perhaps the only one which justifies this paper, was the degree of lightness obtained in the plant.

The Admiralty has always accentuated the need for minimum machinery weights, particularly in small ships; the emergency destroyers built in large numbers during the war had a total machinery weight, including lubricating oil and feed water, shafting, propellers, funnels, uptakes, gratings, but not fuel or spare gear, of about 31 lb./SHP.

The weight allowed for the corresponding complete 8000 SHP. installation in the S.G.B.s was 50 tons—or 14 lb./SHP. The weight actually achieved was just inside this figure, by taking every means of reducing weight available at that time; remembering, however, that decisions were tempered by the knowledge that there was no time for prototype trials and that practically everything had to work "first time out."

Apart from drastic weight-cutting in the actual machinery, every item which had to be included in the total machinery weight was examined to discover if a few pounds could be saved. Even the store pattern engine-room clock was rejected in favor of a lighter model; and the pitch of ladder rungs was opened out!

Wherever possible, castings were eliminated in favor of parts fabricated by welding, and this proved one of the greatest single means of weight saving.

Main Engines.

The boiler output was 80,000 lb. of steam an hour at 400 lb. sq. in. and 700° F., at the stop valve and was fed through a single 5-in. pipe branching to the two sets of main turbines.

The ahead turbines had a Curtis wheel followed by six impulse stages and had a full power speed of 5000 RPM. An astern power of 450 SHP. in each shaft was provided by a single Curtis wheel at the after end of the rotors. The rotors were of the solid "gashed" type.

The dual flow underslung condenser was designed to give a vacuum of 26.5 in. at full power ahead. The shell was fabricated integrally with the lower half of the turbine casing. Longitudinal expansion of the turbine casing was provided for by a vertical plate giving flexibility only in the required direction; collapse of this plate under heavy shock loading was prevented by the absorption, under such circumstances, of a small clearance between more solid members, which then took the load.

The turbine and combined condenser were mounted on feet welded to the bottom of the condenser shell at each end adjacent to the tube plates.

The condenser tubes were of aluminium brass of 19 s.w.g. and the tube plates of cupro-nickel; special Crane packing enabled a reduction to be made in the tube plate thickness, giving a considerable saving in weight. A further saving in weight was obtained by mounting the air ejector coolers in a pocket in the condenser doors, thus avoiding the need for a separate water jacket and piping.

A flexible quill shaft bolted solidly to the rotor spindle passed through the pinion to a claw coupling at the after end, providing flexibility. The pinion was of 3½ per cent nickel steel; the gearwheel was of conventional bolted-up construction, with a hollow spindle. The pitch line speed, tooth loading, materials and other design characteristics of the gearing all had conservative values.

The Michell main thrust block was incorporated in an extension of the fabricated gear case, which was suitably stiffened.

Both sets of engines were controlled by streamline plug type maneuvering valves arranged on the inboard side of each turbine.

From this position one man could keep watch on the four valves. The tachometers were situated on the turbine forward bearing covers driven through gearing from the turbine spindles and readily visible from the control position.

The total weight of the 8000 SHP. main engines of each vessel, including turbines, condensers and gearing, was only 14¾ tons or 4.12 lb./SHP. Messrs. Metropolitan-Vickers were responsible for the design and construction of these parts. These figures, together with the entirely satisfactory running in service, speak for themselves, and must be considered to represent a unique achievement even for a firm with their technical experience and resources.

Boilers.

Preliminary investigation of a number of designs gave a range of weights for the required output of from 20 to 28½ tons. Of these the John Thompson-La Mont and the Foster Wheeler modified D-type boilers were selected as being the two most suitable types for development.

At a very early stage the axiom was laid down that the two designs should be completely interchangeable, so that in the event of damage it would be possible to replace a boiler by one of a different type without rearranging the boiler-room machinery or pipe layouts. Having regard to the marked dissimilarity between the two types selected, it is surprising that it proved possible to meet this requirement and it reflects credit on the firms who co-operated most wholeheartedly.

Apart from the work of opening decks to effect the change, the only alterations involved were the closing lengths of feed and waste steam pipes. The position of the outlet flange of the main steam stop valve was identical for both designs. The uptake was treated as forming an integral part of the boiler.

Design Limitations—In order to reduce the weight of the boiler plant, it was agreed that 72 per cent efficiency would be acceptable for the full power condition. The inlet feed temperature was 193° F. for natural circulation and 222° F. for forced circulation boilers.

The only relaxations from standard stress allowances, etc.,

were that the Admiralty standard drum and pipe stresses could be increased 10 per cent and that flange thicknesses could be reduced 10 per cent. At a later stage, however, boiler-tube thicknesses were reduced two gauges from the normal naval standard.

It was also hoped to fabricate the drums out of a manganese steel which permitted considerably increased stresses and therefore reduction of scantlings, but it was found impossible satisfactorily to weld production plates of this steel, and therefore at a late stage 28/32 boiler quality steel was adopted. This necessitated accepting some higher stresses in parts of the boiler shell, but experience was satisfactory over the period for which these ships remained in service.

John Thompson-La Mont Boiler—This boiler was a development of the conventional La Mont forced circulation boiler. This boiler incorporates an economizer for a large proportion of the convection heating surface; some safeguards were, therefore, necessary to protect the tubes in the event of steam being formed in them. Re-circulation of boiler water was arranged at low powers by means of a differential pressure operated valve. In addition, the flow in the last passes of each element was arranged in an upward direction so that should steam form it would vent freely to the drum.

Circulation Pump—Of the possible designs, that finally adopted was built by Messrs. Mather and Platt incorporating a British Thomson Houston turbine. The duty of this pump was to circulate 480,000 lb./hr. of water at 450° F., and 425 lb. sq. in. suction pressure against a resistance of 30 lb. sq. in, this being achieved for a steam consumption of 2080 lb./hr. The pump was hung from the drum on two suction branches and a steady at the bottom ensured that rolling did not add to the stress load of the suction pipes. In addition, part of the weight of the distribution header was taken on the pump dicharge branch.

The design of the pump gland was particularly successful when compared with the similar fitting in the *Ilex*. In this case the gland was cooled with a small quantity of water from the extractor pump discharge. The leakage through the packing was maintained at a very slight drip, and it was gratifying to find that after twelve months' service the original packing was in good

enough condition to replace after the pump had been opened for examination.

Drum Internals—The internal fittings of the steam and water drum are shown in Fig. 3. All the heating surface including the economizer discharges inside the baffle plates and spills on to the water surface, whence the water passes through perforated baffle

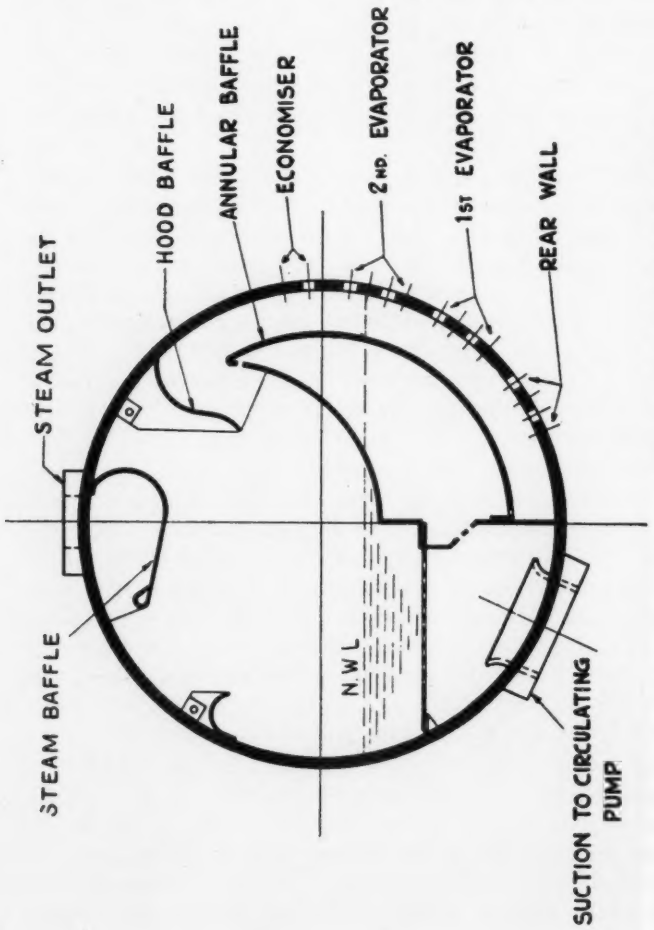


FIG. 3.—ARRANGEMENT OF DRUM INTERNALS OF THE LA MONT S.G.B. BOILER

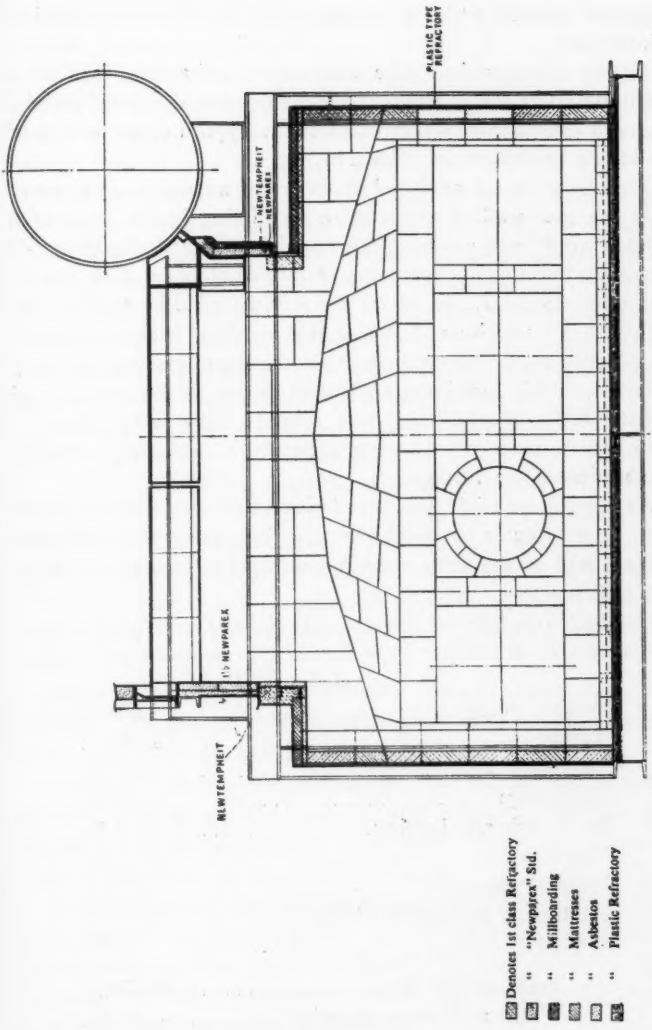


FIG. 4.—TYPICAL ARRANGEMENT OF BRICKWORK AND INSULATION OF THE LA MONT S.G.B. BOILER

plates to pump suctions. The steam is taken out of the drum through a collector. As originally designed, drain pipes were fitted

from the collector to the water space, but these were removed at a later date.

Casing Construction—The amount of refractory brickwork was reduced by the adoption of the water-cooled furnace, thereby saving a considerable amount of weight. Typical constructions of the casing insulation are shown in Fig. 4.

The burners were arranged on an arc of a circle so as to focus near the rear wall. A small refractory lining in the form of a "Dutch oven" was provided near the burners with the object of forming a hot zone to ensure satisfactory combustion. The furnace was rated up to 17.6 lb. of oil per ft.^{3} of CCV., i.e. 0.343×10^6 BTU./ft.^{3} hr., which was less than that obtaining in the shore trials of a highly rated 3-drum boiler, but the flame cooling was very different, in this case being equivalent to that in the conventional 3-drum boiler at about one-half output. This factor, together with others, gave rise to some difficulty in obtaining complete combustion in the furnace.

As a result of trials, the later boilers were fitted with a Dutch oven 1 ft. greater in depth, thereby increasing the refractory surface and at the same time decreasing the volumetric rating.

Boiler Weights—

Original Schedule of Estimated Weights for John Thompson—La Mont Marine Type Forced Circulation Water-Tube Boiler

Welded drum	4,850 lb.
Tubing	10,888 lb.
Steelwork and casings	6,037 lb.
Mountings	1,739 lb.
Brickwork and lagging	6,691 lb.
Headers	2,162 lb.
Integral pipework	1,327 lb.
Circulating pump and turbine	2,184 lb.
Supports	700 lb.
Total weight dry	36,578 lb.
Water and steam capacity	4,095 lb.
Grand total	40,673 lb.
	= 18.15 tons

The first boiler weighed 16¾ tons on completion with the weight of water added.

It should be noted that the original estimate of weight included the use of the manganese steel drum and that the actual weighed weight included the mild steel drum which was some 924 lb. heavier.

Trial Results—On trial the boiler failed to produce its designed performance entirely on account of the inability of the Admiralty oil-burning equipment to complete the combustion of the distillate fuel with the guaranteed excess air figures within the confines of the furnace. Modifications to the oil-burning apparatus and the use of a slightly heavier distillate fuel resulted in some improvement in the combustion, but it was never possible to get the excess air down to the design estimate and keep it there.

Owing to the heavy auxiliary steam consumption in the boat, there was considerable surplus exhaust which was further increased by the exhaust from the circulation pump. The cruising efficiency of the cycle was, therefore, less for the La Mont boiler than for the natural circulation Foster Wheeler boiler, and a two-speed device comprising a Hopkinson uniflow valve with a small constant by-pass orifice of sufficient diameter to provide adequate circulation at the cruising condition was, therefore, fitted to the pump.

Foster Wheeler Boiler—The Foster Wheeler Boiler was a conventional "D" type or 2-drum boiler, modified to use the available space to the best advantage.

External unheated downcomers provided the major proportion of water circulating to the tubes, the side wall being supplied from the bottom drum by means of floor tubes which were protected from furnace heat by refractory bricks.

Drum Internals—The internal fittings of this design consisted of a series of perforated plates, the function of which was to impose some resistance on the release of steam from the water surface, thereby avoiding carry-over in the 33-in. diameter drum on which this design had been based.

The position of the internal feed pipe was changed from the downcast to the riser side as a result of severe water level hunting experienced on trials.

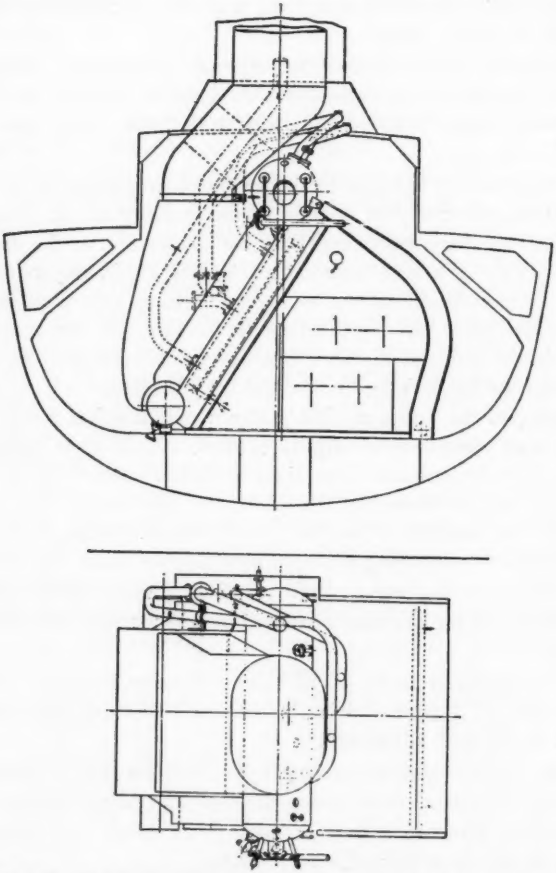


FIG. 5.—GENERAL ARRANGEMENT OF FOSTER WHEELER LIGHT-WEIGHT BOILER IN HULL

Casing Construction—This was the first natural circulation boiler in the Royal Navy for which the casing insulation had been designed to meet a specified target figure. Simple insulation of the type usually fitted required considerable air casing to reduce the surface temperatures to acceptable limits, but by utilizing available insulating materials to the best advantage it was found

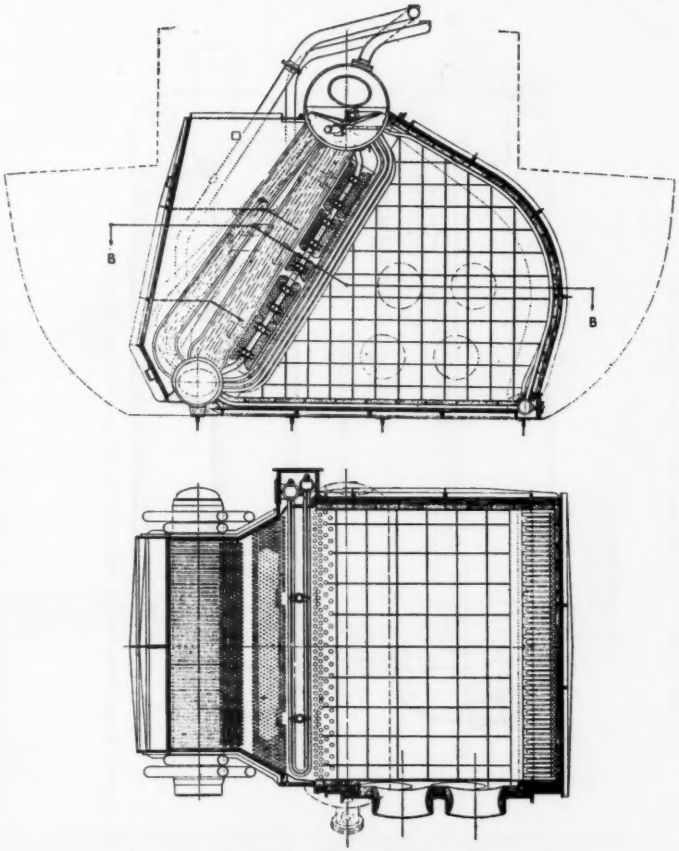


FIG. 6—SECTIONAL ELEVATION AND PLAN OF THE FOSTER WHEELER LIGHT-WEIGHT BOILER

possible not only to dispense with air casings, but also to save weight on the overall casing construction whilst at the same time improving the general standard of heat retention.

Weight of Foster Wheeler Boiler—The progress of weight reduction for the Foster Wheeler boiler is shown in Table I. It should be noted that the first two estimated weights were based on the use of a manganese steel drum and that the final weights were based on a mild steel drum, which was 816 lb. heavier than the original.

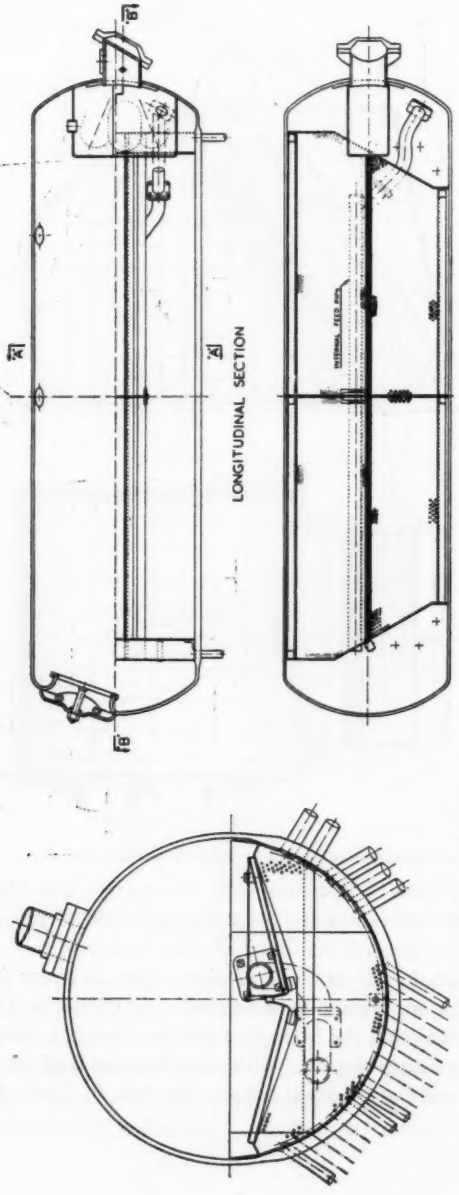


FIG. 7.—ARRANGEMENT OF DRUM INTERNALS OF THE FOSTER WHEELER LIGHT-WEIGHT BOILER

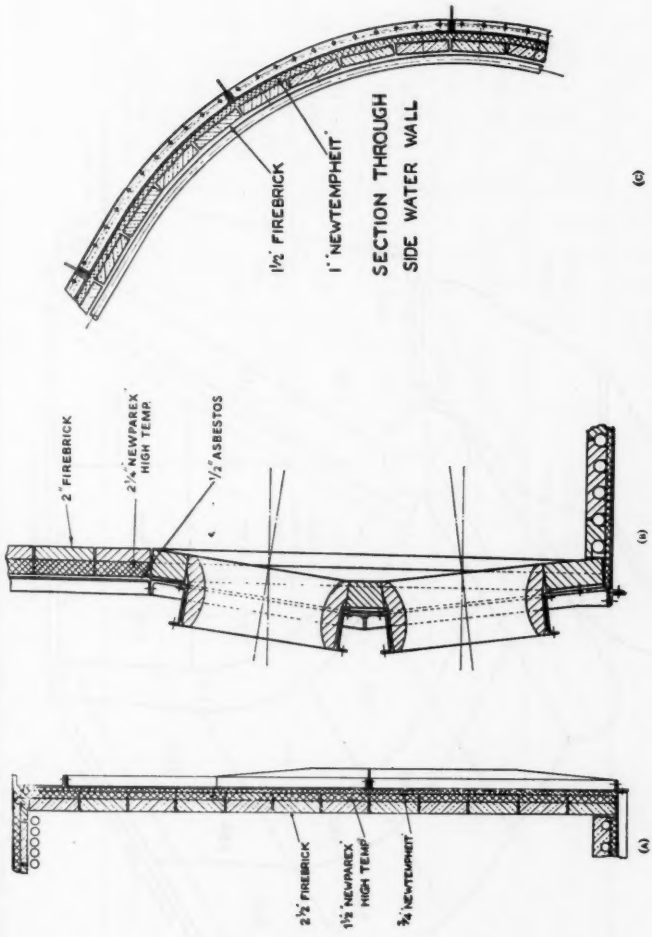


FIG. 8.—THREE SKETCHES OF TYPICAL INSULATION OF FOSTER WHEELER LIGHT-WEIGHT BOILER

It should be noted that the natural circulation boiler was slightly penalized in comparison with the forced circulation boiler, as the use of economisers of a type and design untested by the Royal Navy was not permitted.

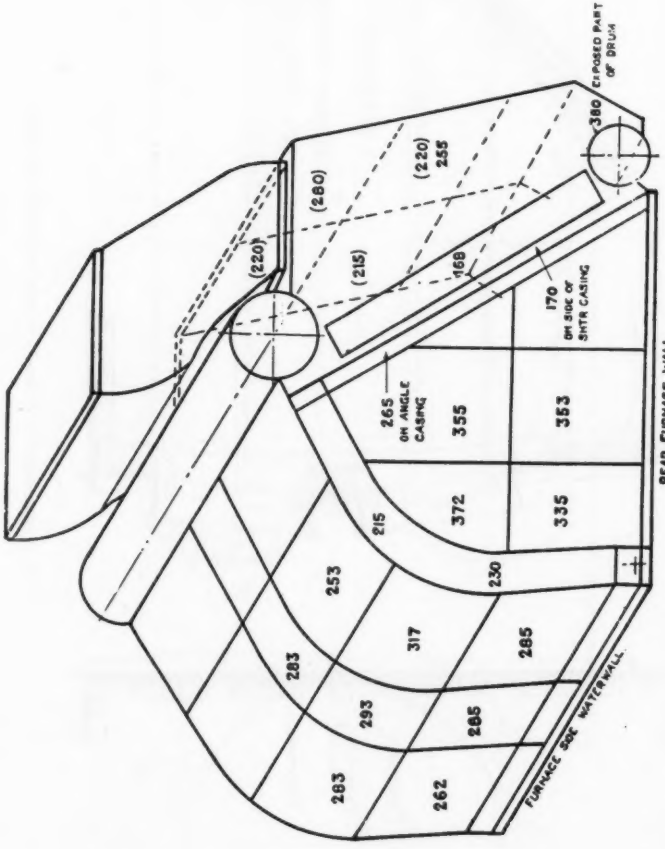


FIG. 9.—ISOMETRIC SKETCH OF FOSTER WHEELER LIGHT-WEIGHT BOILER SHOWING CASING TEMPERATURES MEASURED ON TRIALS

FIG. 9.—ISOMETRIC SKETCH OF FOSTER WHEELER LIGHT-WEIGHT BOILER SHOWING CASING TEMPERATURES MEASURED ON TRIALS

TABLE I

Components	Estimated July 1940	Estimated November 1940	Calculated and weighed April 1941
Steam drum Water drum Tubes Water wall headers	1·55 0·637 6·18 0·149 } 8·516	7·14	3,868 lb. 1,711 lb. 10,601 lb. 1,005 lb. } 7·66
Superheater tubes Superheater headers, etc.	0·808 0·275 } 1·083	1·082	1,490 lb. 994 lb. } 1·11
Brickwork Insulation	3·47 0·915 } 4·385	3·49 0·90 } 4·39	7,340 lb. 1,900 lb. } 4·12
Casings, inc. burners	2·24	2·83	6,125 lb.—2·735
Mountings	1·5	0·723	1,262 lb.—0·564
Dry weight Wet weight	17·7 tons 20·2 tons	16·2 tons 18·5 tons	16·2 tons 18·6 tons

Ratings—Due to its geometry, the furnace of this boiler was more conservatively rated than the La Mont design, the main figures being:—

Oil burned per cu. ft. of combustion chamber: 14.5 lb./ft.^{3} hr., or 0.282×10^6 BTU./ft.^{3} hr.

Oil burned per projected sq. ft. of radiant heat surface: 36.0 lb./ft.^{2} hr.

Less trouble was, therefore, experienced with the combustion, as the flame was not over-cooled.

Control of Feed-Water—Both designs of boiler were fitted with the Weir's modified Steadiflow feed regulator. In order to save weight, the float box of the conventional design was replaced by a tube inserted in the front end of the drum. On this was carried the regulator, the float gear being accessible through an end cover. This design was very successful and only the inability to deal with emergency repairs without shutting the boiler down prevented its wider adoption.

Trial Results—The boiler was fitted with a newly developed open front register which proved very successful. Some wetness was experienced both on trials and in operation, due to the high loading of the drum release area, and it was unfortunate that the stress of war conditions precluded the prolongation of the trials for the development of the internal plate baffles. These baffles are very sensitive to water level, and further modifications to detail were necessary after the boats had been found to trim by the stern.

Experience on Service—The Foster Wheeler boiler was confined to three ships, one of which was sunk at an early stage of its life, so it is not surprising that few troubles were experienced.

The La Mont boiler was in service longer and in more boats, and some troubles were experienced. There were three main causes: firstly, bad combustion; secondly, impure feed-water; and thirdly, action damage.

Defects Caused by Bad Combustion—It is, of course, well known that the flow in an economiser section may be unstable if the heat input is such that steam can be generated. In this design the margin was small and the economiser elements were of necessity arranged perpendicular to the axis of the burners. With the bad combustion conditions in the early boilers, the heat

input to the economiser was increased by the increased excess air and local increases at the back of the boiler due to zoning also occurred. The small margins were, therefore, absorbed and overheating of some economiser elements resulted.

This was satisfactorily overcome by the insertion of inlet nozzles as restrictions in each element, thereby smoothing out the unstable part of the pressure drop-flow curves.

Defects due to Impure Feed-Water—The boiler makers advised chemical feed-water treatment with caustic soda and Calgon. Since the operating personnel lacked experience of such treatment, difficulty in control was envisaged. It was also feared that the treatment might affect adversely the operation of the feed-water regulator and the speed governors of auxiliary machinery if carry-over occurred. The treatment was in fact put into force in the first boat to go on trials, but was abandoned owing to the sluggish operation of the feed regulator, though possibly this was due to other causes. When the boats went into service, therefore, the usual naval practice of treating the boiler water with lime was followed.

The evaporators were very sensitive to operating conditions and other possible causes of sea-water contamination arose due to the peculiar operating requirements of these boats; the boiler densities rose and priming occurred.

Subsequently corrosion of the whole heating surface was found to be active, with particularly marked activity in the region of the superheater outlet.

After attempting to prevent this by greater attention to the supervision, it was finally decided to adopt the United States Navy system using boiler compound, and further trouble from this cause was not experienced.

Defects due to Action Damage—The history of the La Mont boilers under action damage conditions may be regarded as a major triumph.

Two main cases occurred. In the first a bullet hole in one condenser directly contaminated the feed. This accumulated in the boiler, but the ship made port safely, having steamed 3½ hours at a drum pressure of 270 lb. sq. in. and achieved in that time a density of some three times sea water (14,000 grains per

gallon). In the second instance, a tube in the economiser was cut by a cannon shell and an exhaust pipe was holed. The resultant loss of feed-water was made good by direct sea make-up, and once again the ship steamed home safely across the Channel.

The treatment given to bôth of these boilers was to wash through for 12 hours with hot distilled water to remove loose sludge, and then to boil out for 6 hours with caustic soda and sodium triphosphate, and finally to wash through with distilled water. Sample lengths of tubes were then cut out from the more highly rated sections of the boiler for examination, and it was particularly interesting to find that there was no scale deposition on the heating surface. The boiler was then put back into service again.

Propellers and Shafts.

The propeller shafting of 34-38 ton forged steel was stressed to only slightly above the normal Admiralty practice; but every means had to be adopted to reduce the stress in way of the propeller keyways. The keys were of maximum possible effective length, and very shallow. A very high crushing stress was accepted in preference to a high torsional stress in the shaft; very ample radii were allowed at the bottom of the keyways. Although the design was based upon the whole drive being taken by the keys, it was hoped that friction in the propeller cone would in fact do a fair part of the driving.

The shafts were each supported by a single plummer block of the Mitchell self-lubricating type. The propeller brackets were fitted with "Cutless" rubber bearings, and the stern tubes were white metal lined and fitted with Tecalemit lubricators.

In order to reduce the weight of the propeller, the Admiralty stress allowances were increased by between two and three times to the figures used successfully in the destroyer *Ambuscade* built by Yarrow in 1927.

All designs of propellers were 4 ft. 6 in. diameter and manganese bronze carefully finished. In some cases they were dynamically balanced.

Pipes and Bolts.

In normal steam designs Admiralty machinery specifications stipulated that no bolt should be less than $\frac{5}{8}$ in. diameter—a precaution of some age intended to reduce the chance of stretching by over-tightening. Rescission of this regulation was obtained for the steam gunboats, an action which at once allowed the production of new light-weight pipe flange tables; and it also had other far-reaching effects. The abundant saving in weight and material which resulted are shown by the diagrammatic comparison with the standard cruiser flanges at that time. A secondary effect, the greater ease of maintenance in confined spaces due to reduction in spanner sizes, is of interest. Where possible, the flanges were "scalloped," but assembly difficulties limited the extent to which this could be carried out, as once a scalloped flange is secured to a pipe the position of the holes cannot be marked out from the mating flange during erection.

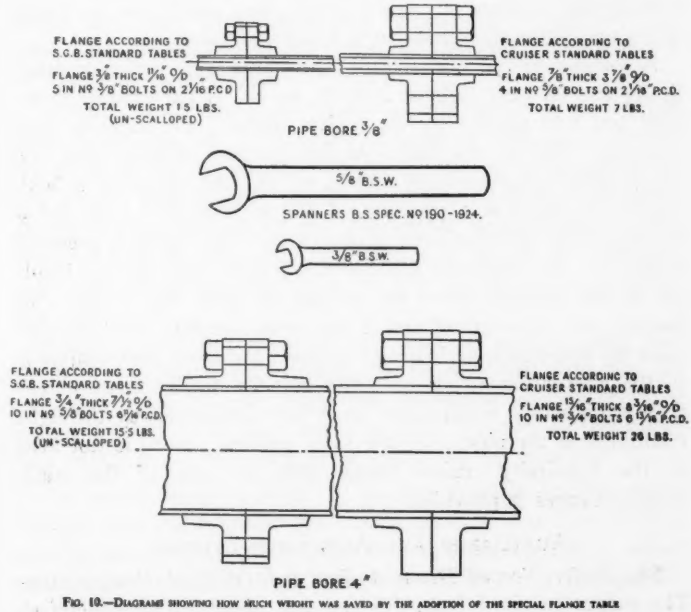

FLANGE ACCORDING TO S.G.B. STANDARD TABLES
FLANGE 3/8" THICK 1 1/16" O/D
5 IN Nº 3/8" BOLTS ON 2 1/16" P.C.D.
TOTAL WEIGHT 15 LBS. (UN-SCALLOPED)
FLANGE ACCORDING TO CRUISER STANDARD TABLES
FLANGE 7/8" THICK 3 7/8" O/D
4 IN Nº 5/8" BOLTS ON 2 1/16" P.C.D.
TOTAL WEIGHT 7 LBS.
PIPE BORE 3/8"
5/8" B.S.W.
SPANNERS B.S. SPEC. Nº 190-1924.
3/8" B.S.W.
FLANGE ACCORDING TO S.G.B. STANDARD TABLES
FLANGE 3/4" THICK 7 1/2" O/D
10 IN Nº 3/8" BOLTS 6 7/16" P.C.D.
TOTAL WEIGHT 15·5 LBS. (UN-SCALLOPED)
FLANGE ACCORDING TO CRUISER STANDARD TABLES
FLANGE 15/16" THICK 8 3/16" O/D
10 IN Nº 3/4" BOLTS @ 6 1/16" P.C.D.
TOTAL WEIGHT 26 LBS.
PIPE BORE 4"

Fig. 10.—Diagrams showing how much weight was saved by the adoption of the special flange table

The now well-known "gramophone" or serrated joint was introduced generally for the first time in naval service in these vessels, but whether the pipe joints were face-to-face or "gramophoned" no case of a leaky joint was ever reported.

Pipe thicknesses were reduced, particularly those of the low pressure system. Six-inch diameter copper pipes carrying closed exhaust were made 0.104 in. wall thickness; smaller sizes were progressively reduced down to 0.041 in. BSF. threads and BSWS. nuts with few exceptions were standardized throughout the plant. Bolts and studs were of 60-ton alloy steel with heat embrittlement resisting qualities; nuts were of mild steel. All low pressure systems had ⅜ in. BSFHT. bolts for every size of flange joints.

Some interesting experiments were carried out by Messrs. Weir to determine if the failing pressures of these light-weight low pressure pipe joints were materially reduced by bending stresses applied to the pipes. The reductions caused by quite considerable bending stresses were in effect small and the joints still gave extremely satisfactory factors of safety.

A number of specimen steam pipe joints were tested hydraulically and all were tight at 3000 lb. sq. in. A particularly stringent test was applied to one typical flange joint made in conformance with the new table. At the works of Messrs. Weir one of the flanges designed for 400 lb. sq. in. was subjected to both water and steam at a pressure of 800 to 1000 lb. sq. in. for a period of six months, being arranged to drain the high pressure steam line. The assembly was horizontal so as to set up any bending action resulting from the mixing of water and steam. No leakage was experienced and it was never touched other than to open the drain cock periodically to drain the water and subject it to the alternating action of water and steam. To some extent these satisfactory results may have been influenced by the light scantlings of the pipes; but the same spotless record is not held by the Admiralty cruiser flange table, in spite of the much heavier flanges it provides.

Auxiliaries and Associated Systems.

The Boiler Forced Draught Fan and Oil-Fuel Equipment—The output required from this fan was about 36,000 cubic feet

of air per minute at 10.5 in. water gauge. In order to reduce weight a fundamentally different design was adopted.

The turbine, developing 85 SHP. at 12,000 RPM., consisted of a single Curtis wheel having a blade p.c.d. of 9 in. The turbine disc itself was of mild steel, with the blades inserted axially and retained by peening over the ends of the slots in the wheel.

The turbine spindle drove the runner shaft through hardened gears giving a reduction ratio of 6.8 : 1. The first set of these gears were straight-toothed, hardened and ground; unfortunately they produced a quite unacceptably high noise level. Facilities for grinding helical teeth were not at that time available to us, but Messrs. Allen hobbed a pair of helical toothed wheels, and hardened them by a process they speedily developed, with so little resulting distortion that these gears, and all those made subsequently by the same method, ran entirely satisfactorily without further treatment.

The turbine spindle ran in plain bearings with a Mitchell type thrust, and was chromium plated in way of the glands; the runner spindle ran in ball and roller bearings; both spindles were of 34-38 ton forged steel.

An over-speed emergency governor was fitted on the driven shaft, which also had at its lower end gearing driving a lubricating oil pump and tachometer. A combined stop valve and emergency valve was fitted to this unit and also to most of the other auxiliaries; the action of the over-speed emergency governor being to close the stop valve itself, avoided the use of a separate valve. A tubular oil cooler was fitted in the air stream from the fan runner.

It was decided, in the interests of simplicity and lightness, that the boiler oil fuel pump should also be combined with the fan unit, and a special gear pump was developed and incorporated with a bevel drive off the runner spindle. The duty of this pump was 16,000 lb. of fuel an hour at 150 lb. sq. in. pressure.

The pump weighed about 10 lb. and actually had an ample margin over all requirements. Some anxiety was felt whether such a pump would prove reliable when drawing fuel from distant tanks, involving fairly high suction heads, but their fear proved ungrounded.

The whole fan and pump unit was suspended from the deck above on spring-loaded shock-proof supports with provision for taking torque reaction. The turbine was connected to its steam supply by a specially flexible length of pipe to avoid the transmission of forces to its light casing; the exhaust pipe was connected to the turbine through a bellows piece for the same reason. This bellows piece was turned from mild steel bar; a prototype was tested for resistance to fatigue in a simple rig devised by the makers, and found entirely satisfactory for the duty.

The unit was arranged so that the quantity of air supplied under service conditions at any fan speed was insufficient to burn all the oil discharged by the pump at that speed; combustion was controlled by by-passing the excess oil through an adjustably spring-loaded valve.

The design of such a small by-pass valve which would hold the pressure constant at the chosen setting irrespective of variation of flow, and within limits of viscosity, proved to be a much more formidable task than might be imagined, but it was successfully accomplished by the makers of the unit.

The weight of the whole of this unit, including the runner, oil fuel pump and all supports and valves, was 845 lb., which can be compared with the total weight of 4500 lb. for the separate fan and oil fuel pump fitted in a *Hunt* class destroyer. Although the oil fuel pump specified conditions for the latter are, it must be admitted, more onerous regarding suction head and governing, the fan duty is considerably less. The steam consumption of the combined unit was quite considerably less than that of the heavier *Hunt* class fan.

This achievement reflects very great credit on the firm, and in particular on Mr. R. C. McLeod, the designer, directly associated with the development.

The upper deck air intake to this fan was required to provide the lowest possible draught loss on the suction side, and also to be so arranged that it would draw in as little spray as possible, and never any green water. These requirements were conflicting, but as a result of air flow experiments conducted by Messrs. Allen a very satisfactory form of intake was obtained, the in-

crease of draught loss due to turning the inlet to face aft being much less than had been expected.

A light oil fuel heater was provided, but not normally used, as the viscosity of the chosen fuels was such that they did not require heating. For lighting up from cold, a foot-operated oil fuel pump was designed and built by Messrs. Allen. To operate this, a man stood with each foot upon a pedal and transferred his weight from one to the other alternately. In this way one man could provide sufficient fuel output to light up and even steam the boiler at low powers for quite considerable periods without undue fatigue.

Main Circulator and Circulating Water System—The necessary head across the circulating water system was normally produced at sea by an inlet scoop. Previous destroyers had had primitive types of scoop, consisting of a convergent nozzle. The steam gunboat had a single scoop of modern type with a divergent nozzle based on information given in a paper read before the Society of Naval Architects and Marine Engineers of America by E. F. Hewins and J. R. Reilly. Although circumstances did not permit a full analysis of the results, the figures obtained showed that the scoop was entirely satisfactory and provided an adequate water supply at all speeds above 4 knots.

To meet standby and astern conditions, a turbine-driven main circulator was provided. The pump body, which, in effect, formed a slight bend in the main circulating water inlet pipe, consisted of a very light gunmental casting with the impeller spindle supported in external ball and roller bearings.

To eliminate the need for bedplates or alternative devices for ensuring the alignment of this pump and its turbine, the drive was taken through a shaft with a Hardy-Spicer flexible coupling at each end. This enabled the turbine to be conveniently hung upon an adjacent bulkhead. The turbine had a three-row radial flow wheel driving through a pair of bevels rather like a motor-car rear axle drive; and a Rover car freewheel was incorporated enabling the impeller to trail and offer the minimum resistance to flow induced by the scoop. The stop valve of the turbine had an extended spindle so that it could be operated conveniently from the main engine throttle position. Turbine and gear casings were welded fabrications of light plate.

The weight of the main circulating pump and turbine complete was 700 lb. compared with rather more than a ton for a similar unit of conventional design.

The design of the circulating water pipes required great care to avoid loss of head and to ensure as far as possible a form which would not lead to erosion troubles. From these two points of view a low water speed was desirable; a suitable compromise had to be struck, therefore, with the need to reduce the pipe sizes as much as possible to reduce the loss of buoyancy and virtual machinery weight.

The upper parts of the condenser were above the load-water-line, and in consequence the discharge pipe had to be swept up inside the hull to ensure that the seawater side of the condenser always ran full. A full-flow main inlet valve was fitted, but this sweeping up of the discharge pipe allowed the omission of a main discharge valve. The actual discharge was below the water-line.

Bellows expansion pieces were not fitted in the circulating water pipes, but, to provide flexibility, thick moulded soft rubber rings were used at every joint in the system; this method saved weight, eliminated causes of eddying and proved quite satisfactory.

The available pressure head across the condenser was applied to circulate cooling water through the lubricating oil cooler and a drain cooler.

A novel type of low resistance circulating water strainer, constructed of straight lengths of streamline section monel metal strip fitted in a rectangular gunmetal frame, was designed by Messrs. Wm. Denny & Bros., and applied to the class.

The Feed-Water System—The condensate ran by gravity from the condenser into a common hotwell tank situated on the centre-line. This arrangement was dictated largely by the need for stable operation under the conditions of heavy rolling which were anticipated. A tank of the required capacity built of steel plate to conventional marine standards was unacceptably heavy, and an alternative was designed by Messrs. Serck Radiators rather on the lines of aircraft practice. It was 6 ft. long, oval section, and built of 14 and 16 gauge brass sheet which avoided the need for

corrosion allowances and permitted methods of fabrication and internal stiffening not possible with steel. The weight was 470 lb., and it proved robust and reliable in service.

The feed tanks were also made of 16 S.W.G. brass, with similar considerable weight saving.

On the forward end of the hotwell tank was fitted a float-operated closed feed control valve specially designed for the service by Messrs. Weir, and weighing only 40 lb. complete. The float chamber and connections were thus avoided. At the forward end of the hotwell tank a gauge glass, illuminated and visible from the control position, was also provided.

The condensate was withdrawn from the hotwell tank by a single extraction pump, and discharged to the feed pump. Both these pumps were specially designed and built by Messrs. Weir. The extractor was driven from what was known as the Combined Unit, which will be mentioned again.

The feed pump was a single-stage pump of the two-bearing type running at 7,500 RPM. direct driven by a turbine with a three-row wheel. The pump delivered 100,000 lb. of water an hour at 525 lb. sq. in. and was pressure governed. Complete, it weighed about 500 lb.; this is about two-fifths the weight for a standard pump of naval type of about the same output.

The feed pump discharge was led to a Weir feed-heater designed with the proportions giving minimum weight for the required heat transfer. In order to save as much as possible of the considerable weights due to the heavy flanges normal on feed-heaters, a light flange and tube plate arrangement was made for test, fitted with the shortest possible tubes, for convenience and economy. From the information gained from this research Messrs. Weir were able to provide a much lightened but still entirely reliable arrangement.

This feed-heater, with an area of 187 sq. ft., weighed, with its lagging, 800 lb.

The Lubricating System—The shaft line of the vessel required that the main gear wheels should be as low as possible in the hull, and this prevented the sumps under them draining by gravity. Consequently, each gear case was provided with its own scavenge pump, discharging to a very light brass lubricating oil tank

specially built by Messrs. Serck. From this tank a single lubricating oil pump took suction and supplied all gear and turbine bearings through a strainer of the self-cleaning type, and a specially light Serck oil cooler. This cooler had no flanges, the water boxes being directly bolted to the tube stack, which could expand freely in the cylinder. The usual Serck safety leakage rings were incorporated.

Individual metering orifices were provided to each bearing.

The twin scavenge pumps and the forced lubrication pump, all of the gear-wheel type, were grouped in one unit—known as the Combined Unit—with the extraction pump, the whole being driven by a geared turbine. This unit, designed and built by Messrs. Weir, was extraordinarily light for the duties it performed, weighing only 6¼ cwt.; it was neat, compact, and efficient.

In order to avoid the possibility of an accumulation of oil in the gear-case sumps, the combined capacity of the twin scavange pumps had to be made greater than that of the lubricating oil pump, so that normally the scavenge pumps handled a proportion of air. Baffle arrangements were provided in the lubricating oil tank and aeration troubles were not met; but an interesting point attributed to this drawing of the engine-room atmosphere into the lubricating system was the high rate of accumulation of water in the lubricating oil, a matter which was rectified by an alteration in the breathing system of the gear cases.

The Dynamo Unit—Electricity for the ship was provided by a direct-driven 10-KW. turbo-generator specially built by Messrs. W. H. Allen, and this unit also provided mechanical power to drive three small hydraulic pumps which worked the power-operated gun mountings and the pump for the hydraulic steering gear. These pumps were all driven by rubber V-belts. The unit, although small, had oil governing gear which was entirely reliable and efficient. The dynamo ran at a speed of 3500 RPM., which was high for naval practice, but which saved weight and proved entirely satisfactory.

The Fire and Bilge Pump—In order to have readily available a fire main service even when the boiler was not steaming, a diesel-driven fire pump was fitted. The unit, designed by Messrs. W. H. Allen, had a centrifugal pump with bilge and fire suctions

placed as low as possible in the ship and driven through a shaft and bevel gears by a Coventry-Victor Cub engine. The pump supplied a fire main running the length of the ship.

The normal naval small-ship practice of arranging the main bilge suction line well above water level was adhered to, and consequently it appeared that this pump could not be used to clear bilges in other compartments unless a comparatively massive air-exhausting device was incorporated, with its attendant disadvantages. This problem was solved, and the weight of the suction line avoided altogether, by fitting an ejector in each compartment operated by the fire-main water supply and discharging directly overboard. This system proved to have a great deal to recommend it.

Also driven by this diesel was a hydraulic pump capable of operating power-worked gun mountings in harbor to deal with short-notice attacks and provide for drilling of gun crews.

The Steering Plant—Hydraulic steering gear giving a rudder head torque of 5 tons-ft. was specially designed by Messrs. Hyland. With a maximum operating pressure of 1000 lb. sq. in., the gear was light and compact, and could put the twin rudders from hard-over to hard-over in 12 seconds. An ingenious arrangement allowed the telemotor control system to operate the rudder rams in an emergency, so that if hydraulic power failed steering by hand power from the same wheel could readily be brought into action. The gear with all its pipes and fittings weighed 11 cwt.

One vessel of the class had an alternative design of gear by Messrs. Hastie.

Distilling Plant—A specially designed evaporator plant with an output of 10 tons per day was produced by Messrs. Weir; 90 per cent of the output was condensed in the main condenser for make-up feed, the remainder was condensed in a separate condenser for domestic purposes.

A small electric motor drove the fresh water and circulating water pumps; and the brine was removed by an ejector operated from the circulating water system.

The weight of the complete plant was about 5 cwt., compared with 30 cwt. for a normal naval design of plant with the same output.

General.

At the end of her career the machinery of S.G.B. No. 9, which acquired fame as H.M.S. *Grey Goose*, was removed and stripped, all important portions being sectioned as necessary in order that a thorough examination could be carried out. This investigation showed that every significant item was still in good condition after five years' war service; the light scantlings having nowhere set a visible limit to future life.

During the design, a working life of five years had been held in mind, although it was not anticipated that the actual life would necessarily be so short except possibly in the case of such items as copper pipes carrying salt water at high velocities; events proved that wear and wastage were almost entirely absent throughout the machinery. It has, of course, to be borne in mind that the actual hours steaming of the vessels were undoubtedly only a fraction of those of major warships, but not all the wastage troubles of the latter are related to the total steaming time. Modern knowledge of protective coatings and corrosion inhibitors could undoubtedly be applied to secure for most light scantling parts a length of life adequate for any class of vessel.

Accessibility of the machinery in these craft was, as may be expected, not everywhere very good; a policy of replacement of units and refit ashore of removed units was intended, and with this in view spare boiler and auxiliaries of every type were provided and stored at the base. Portable deck plates were provided to allow a straight lift for all important units. In service it was found, however, that these facilities were seldom required.

It has been remarked earlier in this paper that between first thoughts on design to actual trials of the completed machinery at sea there was an interval of but fifteen months; we think it will be agreed that there could be few greater tributes to the virility and versatility of the firms taking part than this simple statement.

The greatest enthusiasm was displayed by all those who became connected with the design of the steam gunboat machinery; steam engineers eagerly grasped the chance to show what steam could do as a competitor to internal-combustion engines in the medium-power lightweight field, when released to some extent from the shackles of conventional marine design.

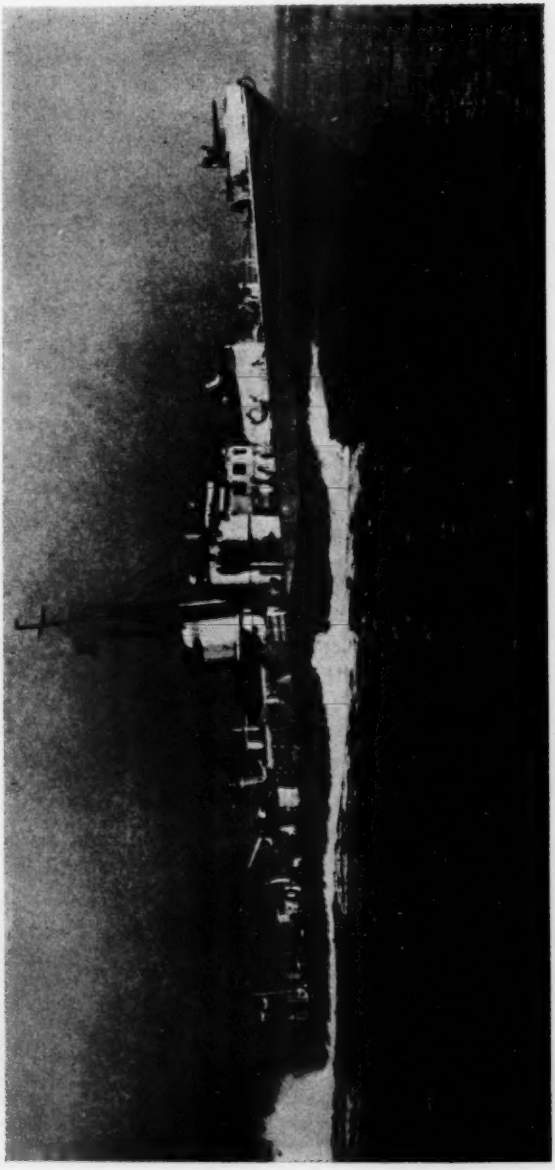


FIGURE 11.—AN S.G.B. STEAMING AT SPEED.

It is unfortunately not possible to refer individually to all those who were associated with this project, but we think that the builders of these craft should be placed on record here:—

Messrs. Wm. Denny & Bros., Ltd., who were the leading firm in the design;

Messrs. Yarrow & Co., Ltd., who completed the first vessel and provided a very great deal of technical assistance, some of it based upon their own designs for destroyers;

Messrs. R. and W. Hawthorn Leslie & Co., Ltd.; and

Messrs. J. Samuel White & Co., Ltd.

This paper is published with the approval of the Lords Commissioners of the Admiralty, but the responsibility for any statements of fact or opinions expressed rest solely with the authors.

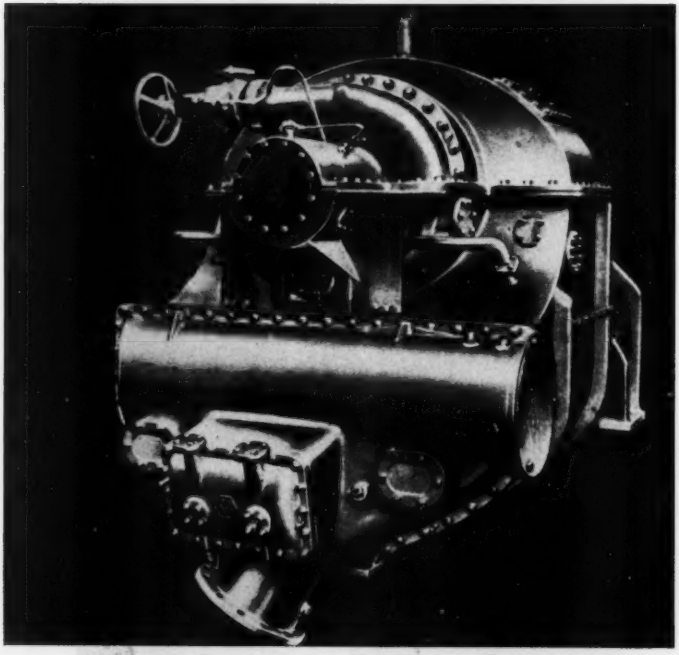


FIGURE 12.—PHOTOGRAPH OF TURBINE AND CONDENSER.

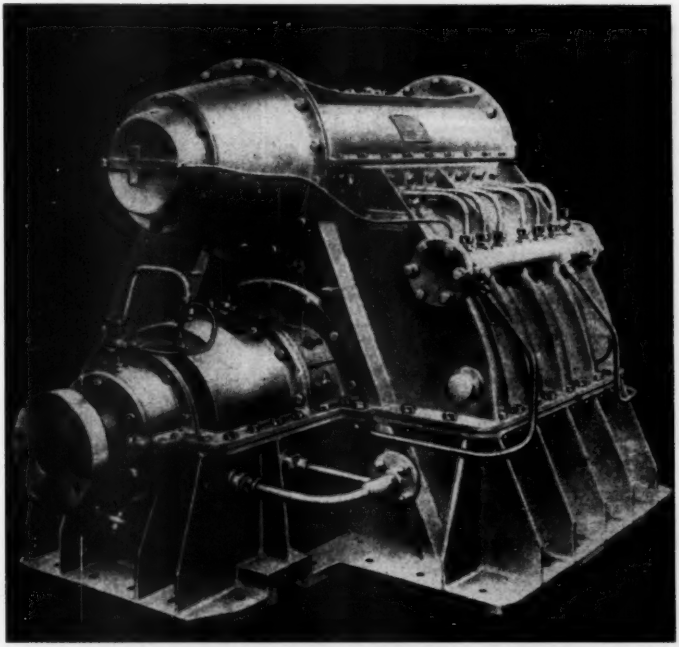


FIGURE 13.—PHOTOGRAPH OF THE GEARCASE.

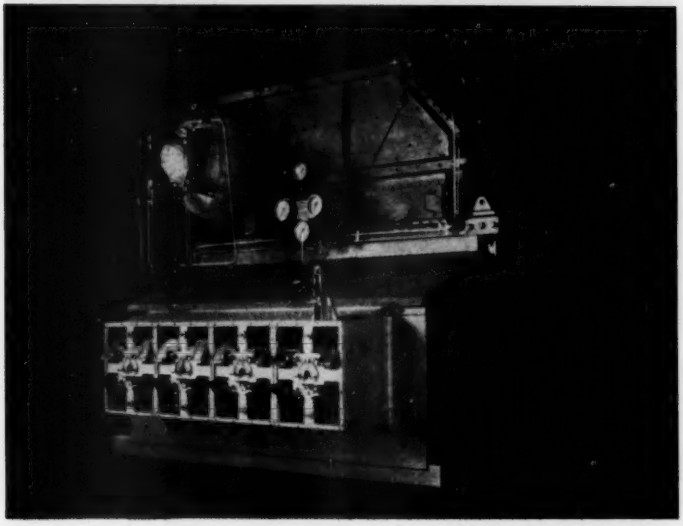


FIGURE 14.—PHOTOGRAPH OF THE LAST LA MONT S.G.B. BOILER WITH MODIFIED OIL BURNING ARRANGEMENTS.

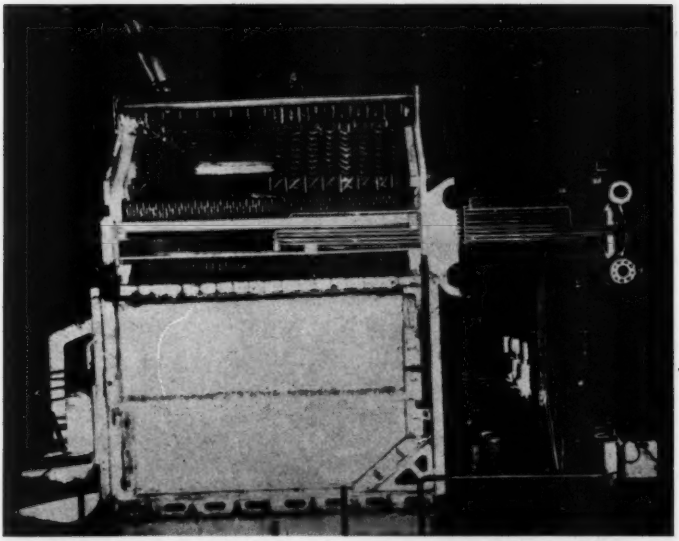


FIGURE 15.—PHOTOGRAPH OF LA MONT S.G.B. BOILER SHOWING SUPER-HEATER PARTIALLY WITHDRAWN. SIDE CASINGS REMOVED TO SHOW ARRANGEMENT OF TUBE BANK AND THE INSULATION ON THE FURNACE WALL.

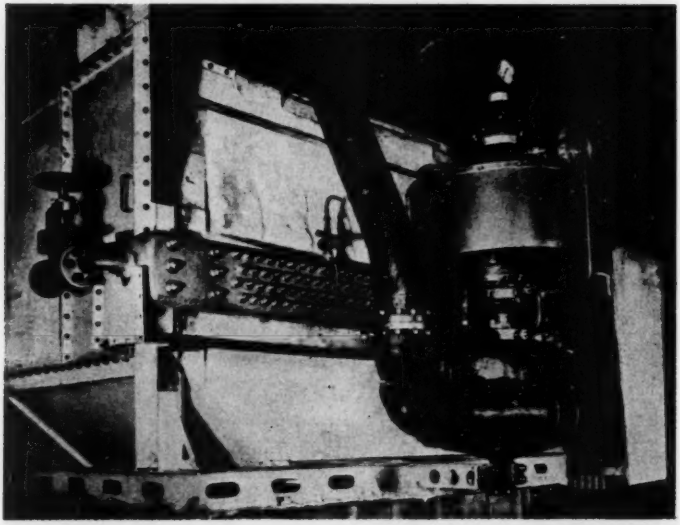


FIGURE 16.—PHOTOGRAPH OF LA MONT S.G.B. BOILER SHOWING THE MATHER & PLATT CIRCULATION PUMP AND DISTRIBUTION HEADER.

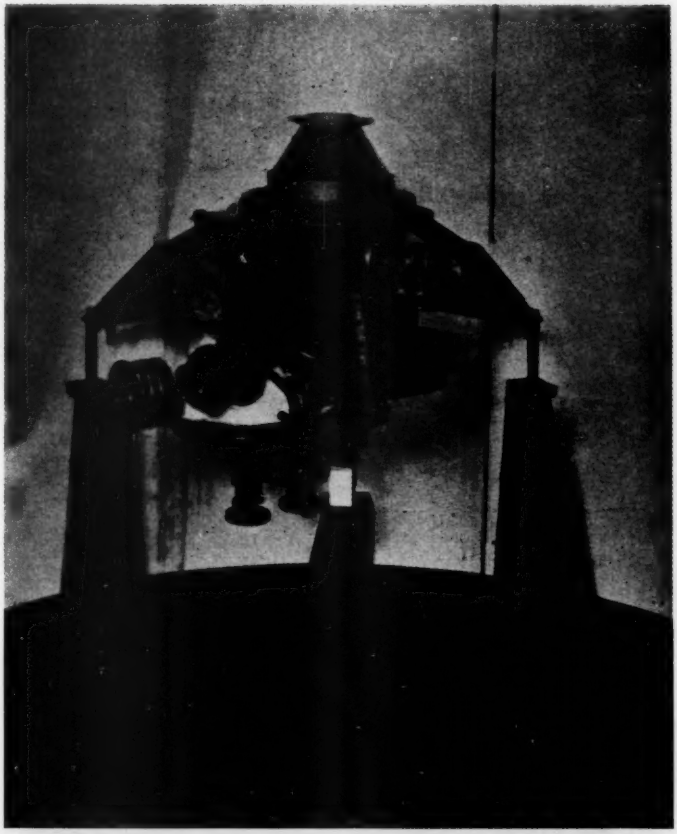


FIGURE 17.—PHOTOGRAPH OF THE TURBINE DRIVING THE FORCED DRAUGHT FAN AND THE OIL FUEL PUMP.

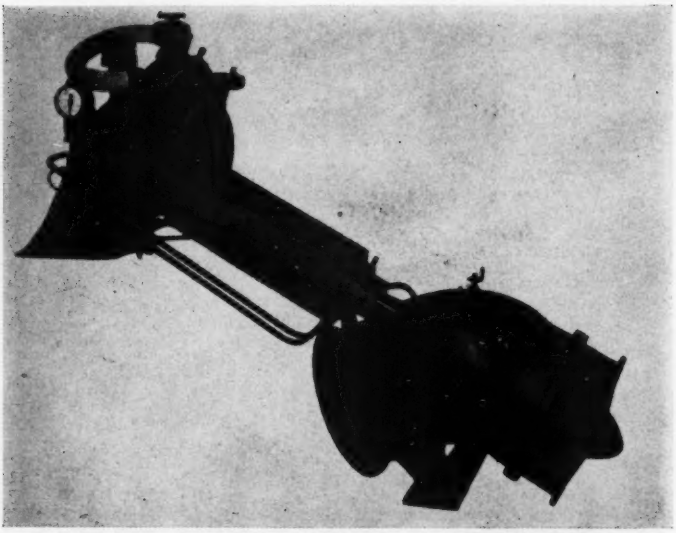


FIGURE 18.—PHOTOGRAPH OF THE ASSEMBLY OF THE AXIAL FLOW CIRCULATING PUMP.

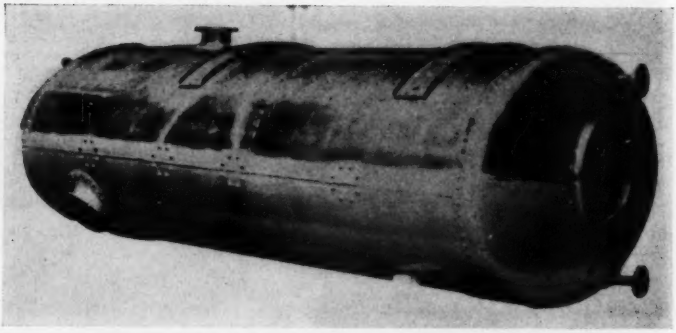


FIGURE 19.—PHOTOGRAPH OF THE HOTWELL TANK.

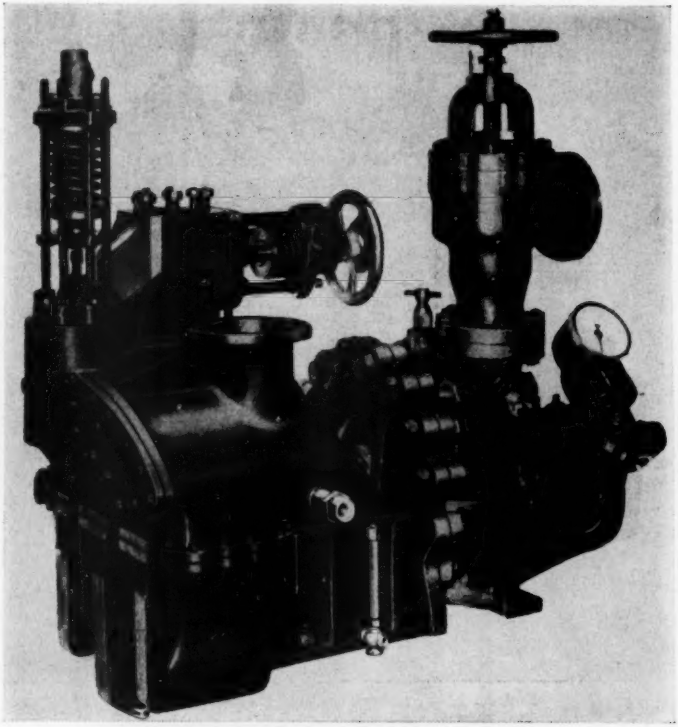


FIGURE 20.—PHOTOGRAPH OF BOILER FEED PUMP.

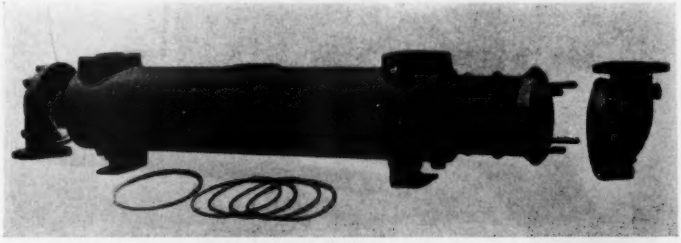


FIGURE 21.—PHOTOGRAPH OF THE SERCK LUBRICATING OIL COOLER.

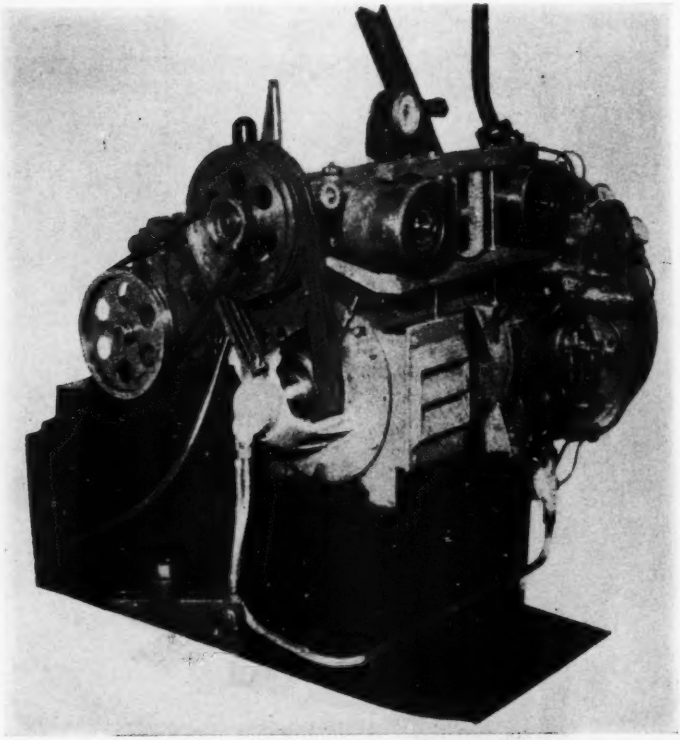


FIGURE 22.—PHOTOGRAPH OF DYNAMO UNIT.

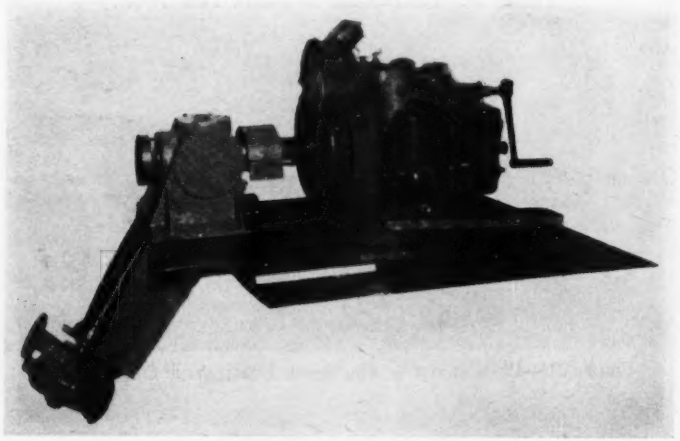


FIGURE 23.—PHOTOGRAPH OF DIESEL DRIVEN FIRE AND BILGE PUMP.

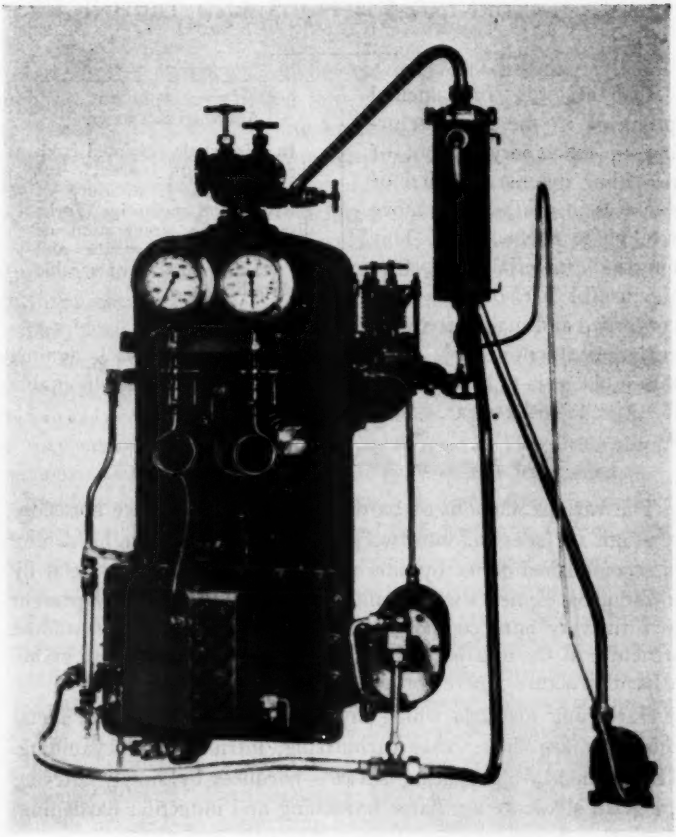


FIGURE 24.—PHOTOGRAPH OF DISTILLING PLANT.

FLAME HARDENING METHODS AND TECHNIQUES.

ACKNOWLEDGEMENT.

In this article reprinted from the "Welding Journal" for October 1948, J. R. Burg of the Baldwin Locomotive Works discusses the preparation of flame hardening apparatus and its use in a number of specific applications.

One of the fundamental problems in the manufacturing of machinery is that of providing against abrasion of surfaces in contact that move relative to one another. This abrasion materially shortens the useful life of the parts concerned and makes frequent replacements necessary. The obvious method of attack against this problem is to harden one of the surfaces in contact to enable it better to resist wear.

TWO METHODS OF SURFACE HARDENING.

The various methods of hardening metals to produce abrasion-resistant surfaces fall into two general categories. The hardening is accompilshed either by altering the chemistry of the metal by introducing elements which unite with substances already present to form very hard compounds, or by rearranging the crystalline structure of the existing metal by some thermal process to create a hard structure where formerly a softer one existed.

Hardening methods which involve a change of surface metal chemistry are three: case carburizing, nitriding and cyaniding. The methods of producing surface hardness by simply altering the grain structure are flame hardening and induction hardening.

Surface hardening by altering surface grain structure is of relatively recent adoption as a manufacturing tool, but has made great strides in the last two decades.

In choosing a method of surface hardening many factors, both technical and economic, must be considered. The degree of hardness, the depth of case, the size and shape of the piece being hardened, are all primary technical factors. Economically the

choice rests on the quantity of parts to be hardened and the facilities available.

Heat and Quench Method.

The "heat and quench" method of surface hardening is supplanting the case-carburizing and cyaniding methods for many applications, its appeal resting on the short time cycle involved relative to carburizing, the simplicity and economy of equipment, and the dependability of the results. Case carburizing requires large furnaces which means a large capital investment and expensive maintenance, plus the expense of a carburizing medium. Further, the time cycle in case carburizing is measured in hours, and may exceed 24 hr.

The "heat and quench" method, by contrast, is a relatively rapid means of obtaining surface hardness. It consists basically of heating the surface of a hardenable steel or cast iron to above its critical temperature and quenching it in water or air to obtain a martensitic structure of high hardness. It is quick, clean, economical and productive of a variety of results as desired.

In addition to the advantages already set forth, flame hardening is very flexible in that it can be used to harden any portion of a surface. To carburize selectively requires elaborate preparation. Also, objects of a wide variety of size and complexity of shape can be flame hardened which might be difficult to case carburize because of furnace capacity. Small quantities can be flame hardened as economically as large, while less-than-furnace loads are uneconomical to run. Finally, flame-hardening equipment can be set up in any convenient location adjacent to other operations, whereas carburizing is usually done in a heat treating department at some distance from machine locations. All the foregoing remarks about case carburizing naturally apply to cyaniding and nitriding as well.

The advantages listed for flame hardening in most cases apply also to induction hardening. There is much overlapping of the fields of usefulness of the two methods and the issue in choosing between them is generally one of economics. The choice is most clearly defined in the case where the quantity of parts to be hardened is very large, and the large original investment for

induction equipment can be amortized successfully; or in the case where the object to be hardened is very large, in which case flame hardening is probably the only applicable method.

Baldwin Applications.

In The Baldwin Locomotive Works, as in the railroad industry in general, flame hardening has been extensively adopted as a means of surface hardening. In our considerations, which will obtain for the majority of shops doing similar work, flame hardening is indicated for most jobs. The apparatus is relatively inexpensive, largely constructed in our own shops, so the original investment is small. The oxygen and acetylene which we use for fuel are readily available in all buildings, and equipment can be set up at any convenient location. Parts can be hardened at stations adjacent to other machine operations. The equipment can be made sufficiently automatic and foolproof so that semiskilled labor is adequate for operating the machines after a short instruction period.

The adoption of flame hardening necessitates, in most cases, the use of a higher carbon steel than formerly employed. There is only a small price premium attached to this, however, and the product almost invariably benefits from the use of a stronger material.

Machine flame hardening is accomplished by one of three methods: progressive, spinning or progressive-spinning. Each method has its applications, and one of the methods will be found applicable to almost any surface hardening problem that may exist. Generally speaking, the progressive method is applied to flat surfaces and large, irregular contours, the spinning method to small cylindrical surfaces and the progressive-spinning method to large cylindrical surfaces.

The machinery used for flame hardening should be designed for a specific job, and should not be general purpose equipment. On this premise it is usually impractical to purchase machinery, which, since it must be custom built, will inevitably prove quite expensive. It is sometimes feasible economically and mechanically to alter equipment built for one job to do another, and in this way the expenditure for new equipment can be avoided. In most plants

the skills and materials are at hand to build very creditable apparatus without resorting to outside sources except for motors and valves and items of that sort.

In addition to being custom built for the job at hand—and, incidentally, this does not mean elaborately or expensively built, but as simply as complete adequacy allows—our equipment is made as fully automatic as the technical and economic aspects of the job admit. Obviously, apparatus built to harden only a few pieces and operated directly under the supervision of a well-informed supervisor need not be made any more elaborate than the bare essentials of the job demand. On the other hand, a large volume of repetitive work justifies the construction of a fairly elaborate machine which will be as fully automatic as possible and leave nothing to the discretion of the operator. Control of all elements connected with the process can be accomplished by mechanical devices if that is desirable. The start, stop, speed of the work or flame head, and the turning on and off of the fuel gases, or any combination of these factors, can be controlled automatically. Full utilization of this inherent potentiality for mechanization has two important results: It makes for exact reproducibility of product and it minimizes the possibility of spoilage due to human error. An operator can be trained in an hour to operate a fully automatic machine, since the task is essentially one of loading and unloading.

The fuel gas regulating equipment used on our flame-hardening equipment is standard, as are the valves and torches. In some cases standard flame heads are used, but generally it has been found advantageous to design heads especially for the job under consideration. Standard flame-hardening heads have been found acceptable for flat surfaces and some cylindrical surfaces, but there are many applications for which they are not adequate.

Adoption of flame hardening for a part formerly case carburized necessitates making the part of a hardenable steel. In choosing a suitable steel we are guided primarily by the surface hardness required. In many applications the surface hardness produced by case carburizing is higher than actually necessary, and a realistic survey of the actual requirements will indicate the possibility of reducing surface hardness and still keeping it within practical

limits. It is patently necessary to use a higher carbon steel than would be used for case carburizing, but this imposes no appreciable penalty. In fact, the higher core strength resulting from the use of higher carbon steel is usually a distinct advantage. It has been our experience that straight carbon steels in the range of 0.40 to 0.60% carbon flame harden readily and produce sufficiently hard surfaces for almost any applications.

Flame hardening has been successfully applied in The Baldwin Locomotive Works to many locomotive parts on which wearing surfaces were formerly obtained by case carburizing or nitriding. In every instance there has been a reduction in cost or an improvement in the product. The list includes reverse links, reverse link blocks, frame stabilizers, guide bars, spring hangers, spring links, hand brake bearings, hand brake collars, turret rollers, tank sprockets and Diesel-engine camshafts.

All these items have one characteristic in common: One or more surfaces are subject to wear and must be hard, and yet hardening all over is undesirable if not absolutely unacceptable. Some of these items will be discussed in detail as being representative of different methods and indicative of the types of objects we think it advisable to flame harden.

Typical Applications.

Our first venture into flame hardening was in connection with sprocket wheels for medium tanks. These wheels, a familiar sight in all tank arsenals during the war, were flame cut from S.A.E. 4140 steel, and flame hardened to resist wear by the tank tracks. The machine devised for this purpose, illustrated in Fig. 1, was similar in principle to many used for this job, but was much simpler in construction than most. The sprocket tooth was heated by being held stationary between two flame heads for a predetermined interval, at which time the sprocket tilted automatically into the water bath over which the apparatus was supported, thus completing the quenching cycle. Pushing down a lever caused the sprocket to index, the next tooth to come up to hardening position, and the torches to light. The entire machine cost less than $500, yet it was almost completely automatic, and compared favorably in volume of production with machines costing many times as much.

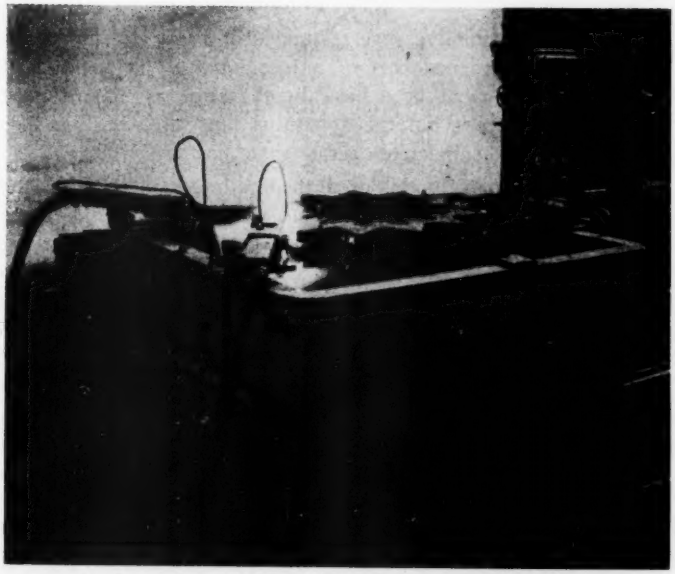


FIGURE 1.—FLAME-HARDENING TANK SPROCKETS.

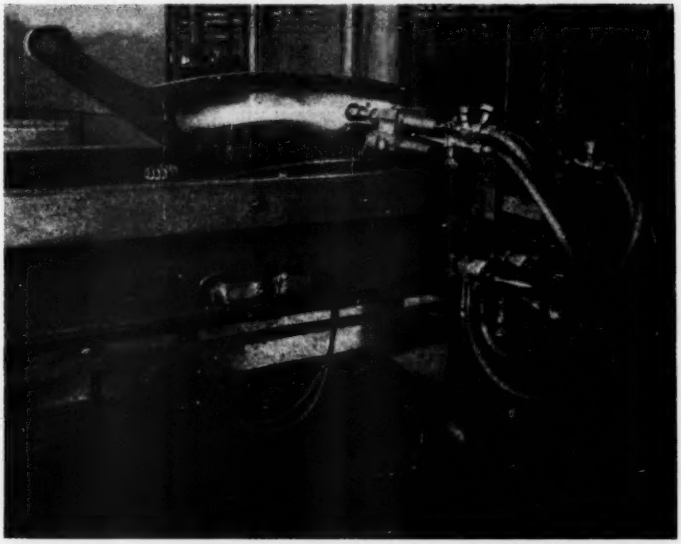


FIGURE 2.—FLAME-HARDENING REVERSE LINK FOR WALSCHAERT TYPE GEAR.

The steam locomotive reverse link is probably the most universally flame-hardened part of a locomotive. Our method of attacking this job has been basically the same as that of most other manufacturers, but the apparatus illustrated in Fig. 2 is more flexible and simpler to operate than most.

The progressive method is used, the link being supported so the wearing surfaces are horizontal and stationary, and the flame and quenching head, one unit in this case, being driven past the surface by a chain and sprocket arrangement. There are two somewhat unique features of this apparatus. The first concerns the manner of driving the flame head. The head assembly runs on rollers along the edge of the tank, propelled by a sprocket engaging a chain which runs horizontally just below the edge of the tank. This sprocket turns as an idler when the machine is started until a dog on the sprocket engages the chain, at which time the head assembly starts to move and the hardening process begins. By altering this idling period, any desired interval of preheat may be obtained. Consequently, the operator's judgment as to proper preheat does not enter into the process. At the end of travel of the flame head, limit switches cut off the gas supply, the acetylene first to prevent flashback and then the oxygen. A static counterbalance automatically returns the machine to the start position and resets the preheat timer.

The other feature of this machine which deserves mention is the method of reproducing the arc of the reverse link surface. A strip of spring steel is used, supported horizontally at the center and pulled down into an arc by studs on each end. This strip serves as a template to guide the flame head. Since it is adjustable as to radius of curvature it can reproduce any reverse link arc, or can be allowed to spring up straight to permit the hardening of plane surfaces such as motor suspension guide bars. The machine is thus capable of a large variety of work and is very simple in operation, nothing being left to the operator's judgment. He has only to load and unload it, and push the starting button.

Another interesting application of flame hardening in our plant arose from the necessity of obtaining wearing surfaces on the shoulders of spool-type rollers for large gun mounts. Again, the hardened area must be only a portion of the total surface and

must be controlled between the rather severe limits 45-48 Rockwell C. The geometry of this piece obviously made it well adapted to the spinning method of flame hardening, therefore, apparatus was designed on that basis. Since the roller was of large mass it was impractical to quench it by dropping into a bath as is often done on smaller pieces. The alternative was to heat and quench the spool without changing its position. This method was used, the spool being spun in the vertical position until proper surface temperature was obtained, at which time the flames were shut off and the quench applied.

The surfaces to be hardened were on the sloping shoulders of the spool. To get the heating and quenching heads sufficiently close to the sloping surface it was necessary that they extend inside the bell ends of the spool, so the machine could be loaded and unloaded only by providing some means of swinging the heating and quenching heads out of the way. This was done by pivoting each head at its support and attaching a sprocket to the vertical shaft on which each turned. The sprockets all engaged a chain that ran horizontally around the machine and could be moved by a hand lever. Thus, a movement of the lever swung all the heads out of the way to allow the spool to be lifted off its turntable. These details may be seen in Figs. 3 and 4.

The quenching medium in this instance was air. It was introduced in specially constructed heads similar to the flame heads and placed alternately with the flame heads around the job. Two problems arise in connection with the use of air as a quenching medium, those of cleanliness and temperature. Ordinary compressed air may carry particles which will obstruct the orifices in a quenching head and alter the effectiveness of the quench. To overcome this difficulty it was necessary to put a strainer in the line. The temperature of the air varied considerably with atmospheric temperature and produced variations in results until an ice bath was provided through which the air was passed to assure uniform temperature. This machine was also made completely automatic, the cycle being controlled entirely by electrical means.

The most ambitious flame hardening project undertaken to date in our organization is that of hardening the wearing surfaces on Diesel-engine camshafts. The cams and bearings on these shafts

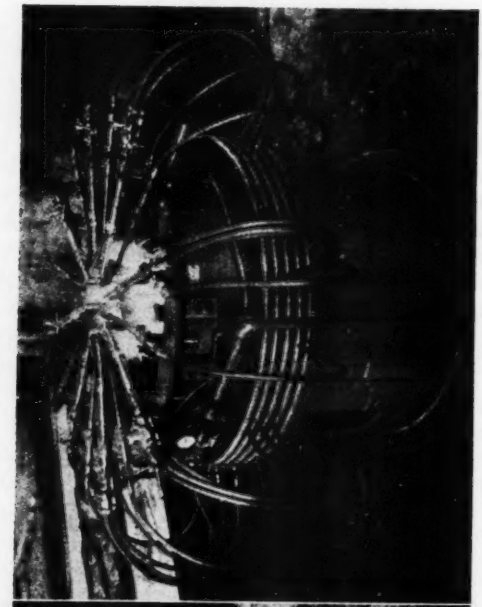


FIGURE 4.—TURRET ROLLER HARDENING APPARATUS IN OPERATION.

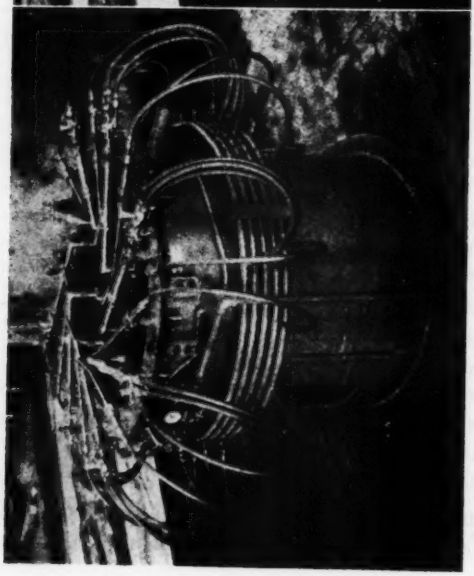


FIGURE 3.—TURRET ROLLER HARDENING APPARATUS IN LOADING POSITION.

have been hardened by case carburizing in the past, a method which produces a satisfactory result but which is time consuming and limited as to production by the capacity of available furnaces. When the need for more camshafts became imperative, means of hardening were sought which would prove more economical than further investment in furnace equipment. It was decided that flame hardening presented the most attractive potentialities, and a development program was initiated to determine the practicability of that process for the application at hand.

There seemed to be three possible approaches to the problem: spinning, progressive hardening around the periphery of the cams or progressive hardening transversely across the cams. Each was given consideration, and apparatus of a temporary nature was set up to try the spinning method, which at first seemed most promising. It was quickly discovered that the heat distribution was very uneven in this method, the highest parts of the cams becoming excessively hot before the base circle area was up to quenching temperature.

Further investigation into other variations of mechanical arrangements led gradually to the belief that progressive hardening around the periphery of the cams offered more promise than any other method. Accordingly, it was decided to build a machine for hardening in this manner.

There is one difficulty of outstanding importance in the implementation of this method. Any cam, because of its unsymmetric shape, has a varying surface speed with respect to a fixed point outside the cam. Since uniform flame hardening depends on constant surface speed relative to the heat source, a method of reconciling these differences, at least partially, had to be devised. Varying the rotational speed of the camshaft seemed to require too intricate a mechanical arrangement.

Another difficulty inherent in any progressive hardening cycle concerns the band of metal, hardened at the start of the cycle, which is reheated but not quenched as the cycle is completed. The result is an area of varying width which is not hardened as completely as the adjacent metal.

The machine which finally evolved as the result of consideration of all these problems is shown in Fig. 5. It is similar in

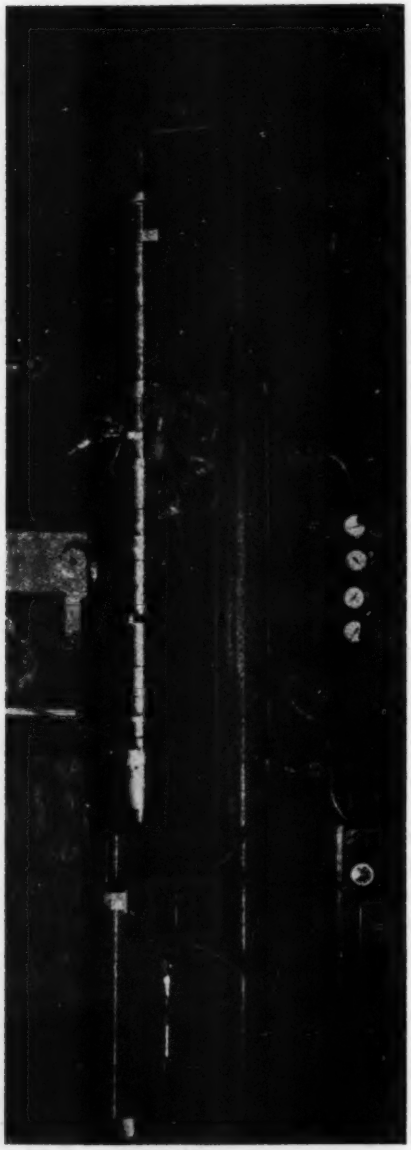


FIGURE 5.—DIESEL ENGINE CAMSHAFT SECTION IN FLAME-HARDENING MACHINE.

FIGURE 6.—ELLIPTIC SPRING HANGER BEING FLAME HARDENED.

FIGURE 7.—MOTOR SUSPENSION GUIDE BAR BEING FLAME HARDENED.

appearance and function to an engine lathe. The shaft is supported between centers and rotated at constant speed. A set of flame heads, mounted in a turret arrangement to facilitate switching from one to another, takes the place of the tool rest assembly. This flame turret can be moved by means of a rack and pinion along the length of the shaft, so as to harden all cams and journal bearings in one shaft setup. On this flame head assembly, back of the shaft, is a set of master cams which controls the movement of the flame heads to maintain any predetermined distance from flame to work during rotation. The hardening is accomplished by rotating the camshaft past the flame heads and quenching head, in this case separate units, and hardening in a progressive manner.

The problem of unequal increments of heat due to irregular surface speeds is answered by regulating flame-to-surface distances with respect to peripheral speeds to keep the unit heat input practically constant. This flame-to-surface distance adjustment is simply a matter of altering the contours of the master cams.

The difficulty arising from the overlap at the starting point, as pointed out, is inherent in the process. Its effect can be minimized by the use of some sort of shield which protects the hardened area from subsequent drawing. Several types of shield and several schemes for positioning them were tried and discarded. The shield must be watercooled, and when it is introduced into the flame area it tends to absorb heat that would otherwise go into the cam surface, and to upset the established thermal balance. Because of the lack of symmetry of the cams it is impractical to try to keep the shield in position during the entire cycle. Finally a cam arrangement was devised to push the shield into place after 270° of rotation of the shaft. The heat balance was preserved by a precise adjustment of rate of flow of cooling water in the shield.

This machine has been in service only a short time, but has proved mechanically sound and promises to be an economical and completely adequate instrument for flame hardening.

Conclusions.

We at Baldwin's hold no brief for flame hardening as a universal method of obtaining hardened surfaces. There are some instances where its use is definitely less desirable than some other method. Flame hardening, where it is technically feasible, has the distinct advantage of lending itself to a wide range of complexity of apparatus, with an accompanying range of initial investment. It can be made very simply and cheaply, or very elaborately and expensively, depending on the requirements of the problem. In passing it might be well to say that an expensive flame-hardening machine would be inexpensive compared to induction-hardening apparatus or to a carburizing furnace. The investment in equipment for flame hardening is relatively small. The major part of the cost is in the development work, which is under close control if it is done within the plant, and the expenditure can be regulated in accordance with the requirements of the job.

As stated before, our belief is that the equipment should be made as nearly automatic as possible. As with other parts of the machine, the cost of equipment for mechanizing operations is relatively small, and reasonable development costs should be the primary criterion in deciding how completely to eliminate the human element. In every case in our experience completely automatic controls have returned their value many times in identity of results and volume of production.

Our belief in the value of flame hardening as a production tool is evidenced by the variety and multiplicity of parts which have been and are now being hardened in our plant.

METHODS OF LUBRICATING HIGH SPEED BALL BEARINGS.

ACKNOWLEDGEMENT.

This article by H. F. Brewster of the Texas Co. is reprinted from "Machinery" for October 1948. It discusses various methods of lubricating high speed ball bearings and their effectiveness.

Ball bearings, unlike plain bearings, do not fail by reason of wear or abrasion—a change occurs in the surface of either the balls or races (or both) to cause flaking or spalling off of minute particles. Lubrication, which keeps the rotating surfaces continually wet with a film of oil, reduces the spot temperature condition which would otherwise lead to metal fatigue. Obviously, it is most important to prevent metal fatigue in a high-speed bearing where hundreds of millions of revolutions are required between overhaul periods, or in some cases, even between shutdown periods. Oil is the ultimate lubricant, regardless of whether the lubricating film is developed by some means of dispersing fluid oil or whether the required oily component for the film is "bled" from the soap base of a grease which is so applied as to be in intimate contact with the rotating parts of the bearings.

Lubrication at the load-carrying surfaces which involve rolling contact, in the opinion of some authorities, is primarily for corrosion prevention. Any sliding velocity which occurs, due to the finite contact ellipse, probably is accompanied by such high contact pressures that oil film action seems unlikely. The major function of the lubricant in a high-speed ball bearing would seem to be to minimize friction between the balls and the ball separator.

Most effectual lubrication of a high-speed bearing by means of oil is attained when the volume of oil circulated over the bearing parts is controlled so that just a sufficient amount is applied to wet the rolling surfaces. Obviously, an excess of oil, such as

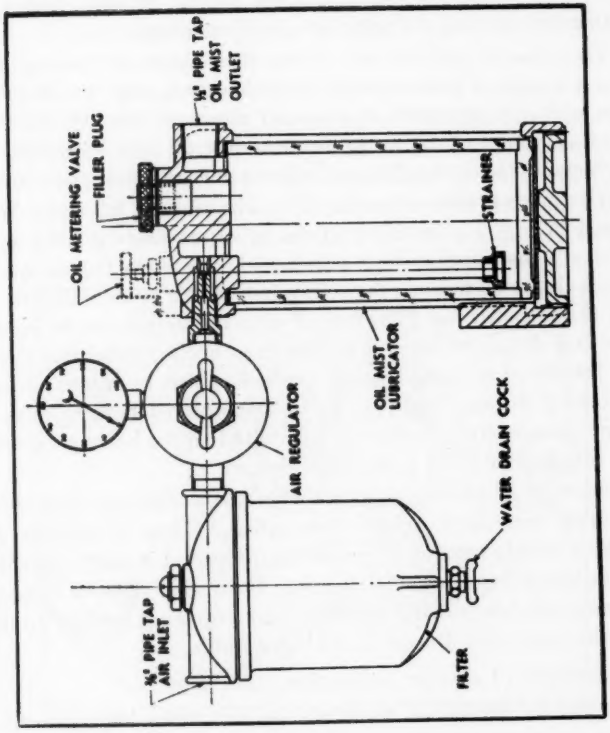


Fig. 1. Diagram of an SKF Oil-mist Lubricator which Supplies a Fog of Lubricant to High-speed Ball Bearings by Injecting Oil into a Stream of Low-pressure Air

would result from flood lubrication, might seriously affect the maximum speed of the rotor due to oil churning.

The oil-mist lubricator is a device that injects oil drop by drop into a stream of low-pressure air which breaks up the oil into a fine mist. A regulator is provided ahead of the lubricator to accurately control the air pressure admitted to the lubricator. A high-quality filter should also be inserted in the line ahead of the lubricator to provide clean dry air. The oil-mist lubricator illustrated in Fig. 1 produces particles of oil so finely divided as to become a fog. In operation, a vacuum is created by the air stream which draws oil through the orifice into the air stream, thus forming the so-called fog. The rate of oil consumption can be limited to a few drops per hour. This fog is so finely divided that it can be distributed to many bearing locations. Also, by means of mist producing fittings, this fog can be transformed back to mist or spray or even drops, whatever is required on the bearing location. All this is done from a single lubricator.

This type of lubrication assembly is widely used for high-speed grinding machinery spindle lubrication. Since it requires air under a certain amount of pressure, obviously it is most applicable to stationary machines. The heading illustration shows a Norgren oil-mist lubricator with air filter and regulator applied to the spindle head of a Bryant chucking grinder.

Advocates of oil-mist lubrication claim that:

1. The air portion of the air-oil fog provides a cooling medium which maintains uniform spindle temperature.

2. By permitting the fog to continue flowing for a short time after shutdown, the creation of a partial vacuum within the spindle housing as it cools down is avoided, and harmful grit will not be drawn into the housing.

3. The particles of oil penetrate the hard core of air surrounding the rapidly rotating bearing to lubricate it without causing uneven rotation.

4. Only new oil is ever carried to the surfaces of the bearing.

5. Air pressure slightly above atmospheric is maintained inside of the spindle housing to prevent entrance of foreign matter, such as abrasives and coolants.

Oil-mist lubrication is not measured lubrication in the true sense of the word, although the procedures perfected provide for very accurate control in terms of drops of oil delivered by the oiler or the wick. Aircraft designers, in seeking for a satisfactory method of lubrication for the high-speed ball bearings of cabin refrigeration and pressurization equipment, have adopted wick lubrication. The desired oil mist is obtained by submerging one end of a wick in an oil reservoir, the other end being in contact with the high-speed rotating shaft, as shown in Fig. 2. At this

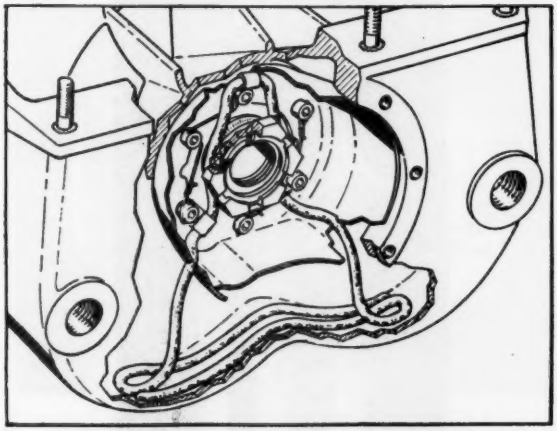


Fig. 2. An Oil-wick Lubricated Installation for an Aircraft Cabin Compressor

point a spray of oil is thrown off. Air draft draws this spray into the bearings; no attempt is made to throw the spray directly into the bearings.

The required air draft is developed by a slinger adjacent to the bearings on the end opposite the wick contact point. Suitable grooves convert this slinger into an effective air pump. When this device goes into operation at high speed, a stable oil mist fills the entire bearing space, and because it contacts all parts of the bearing, it develops a very dependable means of lubrication. No attempt is made to use the oil as a coolant, the desired cooling

being developed by conduction through direct contact with the circulated air. This is a highly effective way of cooling when the heat rejection rates of the bearings permit of air or conduction cooling. When this is not sufficient, oil cooling jets can be used.

Research has determined that the effective oil pumping capacity of such a wick is surprisingly high. The rate of oil flow through a ⅛-inch felt wick in contact with a high-speed shaft is as much as 20 cubic centimeters per hour, by actual observation.

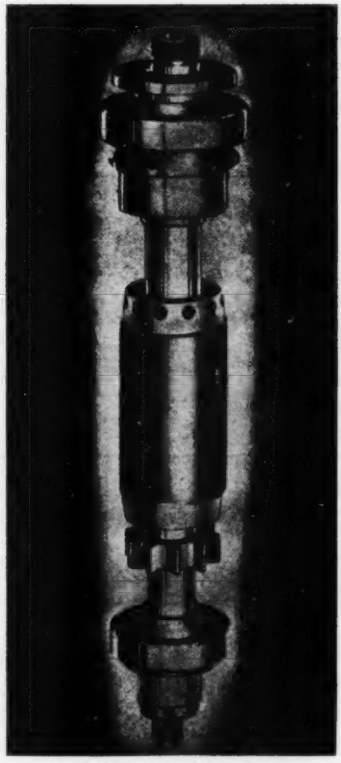


Fig. 3. Whitney Woodworking Shaper Assembly, Showing the Over-size Spindle, Heavy-duty Bearings, Fan, and Oil Slingers

Some atomization of oil is also attained when high-speed bearings are to be lubricated by attaching oil slingers to the shafts. As the latter rotate, the slingers, which dip into an oil reservoir below, carry a continuous film up into the upper part of the assembly, where it is thrown off. An atomizing effect results as oil passes through the bearings. Design of the bearings includes apertures of varying size and number which promote the atomizing effect. Examples of this method of lubricating bearings are shown in Figs. 3 and 4.

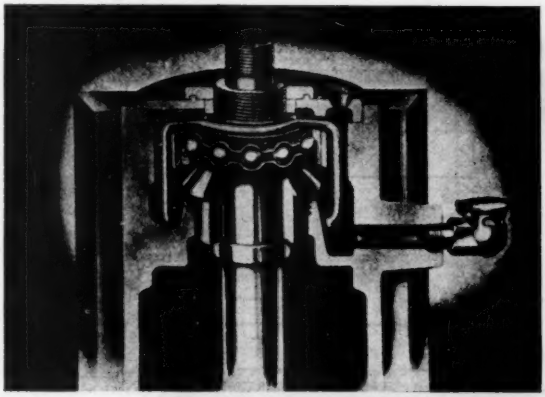


Fig. 4. Whitney Oil Lubrication System, Showing Oil Feed to Ball Bearings by Wicks and Oil Slinger, which Creates a Fine Mist of Oil

Metered mist is another interesting method devised for maintaining dependable oil lubrication of high-speed spindle bearings, especially for woodworking machinery. Metered-mist lubrication employs the centrifugal force developed by spindle rotation as the power for maintaining the oil feed. Fig. 5 illustrates the principle of operation.

As shown at *A*, the hollow spindle is the oil reservoir. Air is trapped in the upper part by the feed-tube, preventing over-filling. As the spindle turns, centrifugal force drives the oil against the walls of the reservoir (see view *B*). The air that was trapped forms the core extending upward into the feed-tube, thus prevent-

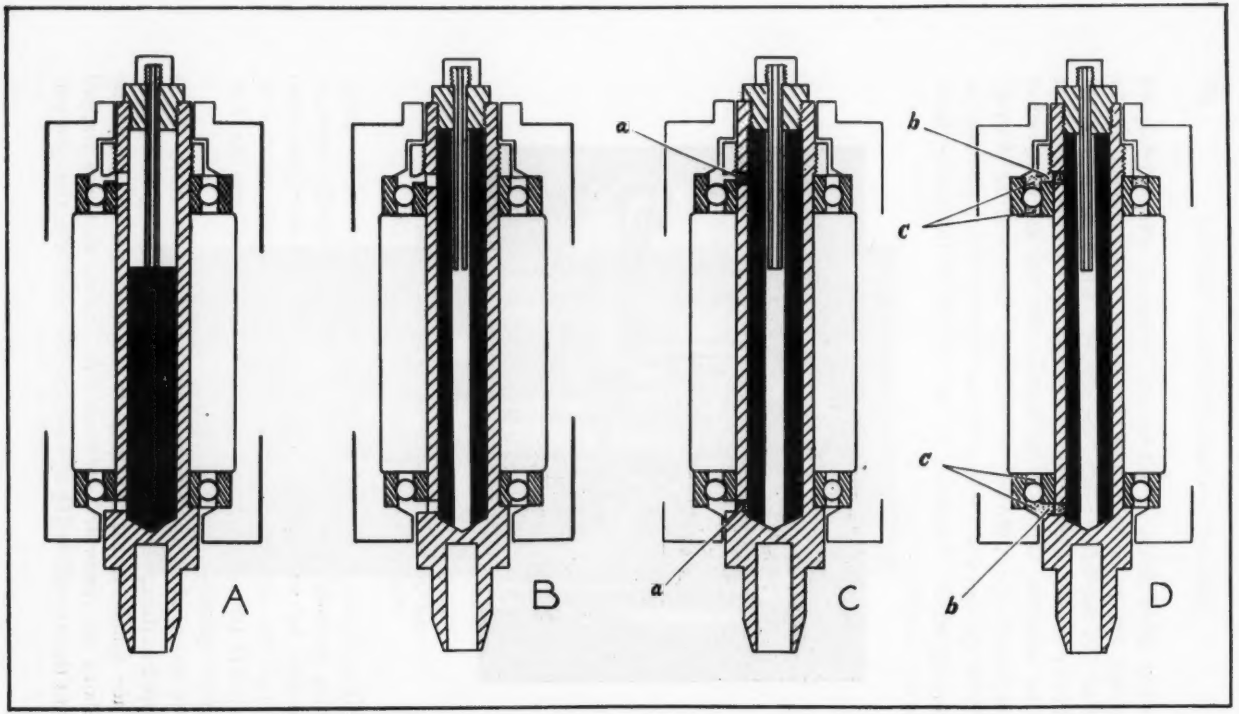


Fig. 5. Diagrammatic Illustration of the Manner in which Metered-mist Lubricating Devices Work. The Hollow Spindle Serves as a Reservoir for the Oil. Metering is Attained by Centrifugal Force which Drives the Oil through a Metering Element of Predetermined Porosity

ing oil leakage. The only points for oil escape, as seen at *C*, are through metering elements *a*. Being of predetermined porosity these permit only the desired amount of oil to pass in a given time. They also break up the oil into mist-like particles. As the oil leaves the metering elements centrifugal force carries it through spaces *b* into bearing chambers *c*, as indicated at *D*. Metered-mist oiling is practicable in any plane.

To provide a quickly available source of oil for high-speed motor spindle bearings during the first few seconds of operation, a unique priming cartridge has been developed for use with a metered-mist system. The priming cartridge is shown in Fig. 6,

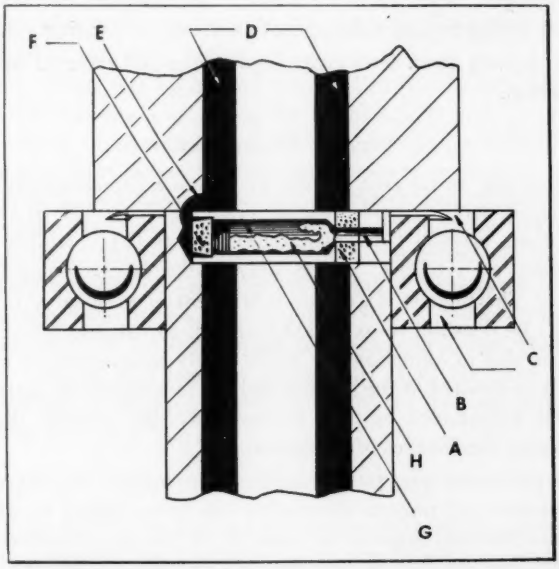


Fig. 6. The Onsrud Priming Cartridge Provides a Quickly Available Source of Oil for High-speed Motor Bearing Spindles during the First Few Seconds of Operation

where *A* is the porous metering plug; *B* the open feed channel; *C* the bearing chambers; *D* lubricant on the walls of the reservoir; *E* the wall cutaway; and *F* a porous plug. The direction of

Work. The Hollow Spindle Serves as a Reservoir for the Oil. Metering is Attained by Centrifugal Force which Drives the Oil through a Metering Element of Predetermined Porosity

lubricant through this plug is toward the axis of rotation. *G* is the priming cartridge. When rotation ceases, the priming wick *H* absorbs the lubricant and holds it for recirculation when the spindle begins to rotate again.

This provision for auxiliary lubrication during the start-up period is very desirable where the unit during shut-down may develop enough heat to evaporate the existing oil film from the spindle bearings. High-cycle electric motors running for lengthy periods under severe operating conditions, when stopped for a few minutes, can become so hot as to cause their bearings to run dry until a new oil film is built up, unless means for oil priming are employed. Two or three drops of oil only are necessary to develop complete film coverage of each set of bearing elements and to protect them until centrifugal force and metered mist go into action.

Type of Oil to Use.

The oil for an oil-mist system should be a comparatively low-viscosity mineral (petroleum) oil (ranging from 100 to around 350 seconds Saybolt Universal at 100 degrees F.). Stability is as important as viscosity, which only indicates the comparative fluidity. The highest degree of refinement is necessary, for the oil must be resistant to oxidation and gum formation. Conditions are favorable to oxidation due to the finely atomized state of the oil as it is directed to the bearing and the fact that each particle is virtually surrounded by air. Furthermore, the warmer the air, the greater the chances of oxidation.

The petroleum chemist has perfected a method of improving the resistance of mineral lubricating oils to oxidation by adding materials that will improve the stability of the oil. Additives also are available which make mineral oils capable of forming rust-resistant films on steels. It is advisable to consider seriously an oil of this type for use in an oil-mist system, particularly if moisture conditions are extreme and machines such as high-speed grinders are involved, which are subject to periodic shut-down. Condensation under such conditions could do serious damage to spindle bearings unless the oil is fortified to prevent rust.

High-speed ball bearings in gas turbines and jet-propulsion

engines carry thrust loads of such magnitude that considerable heat is generated in the bearings due to the coefficient of friction. Obviously, this heat must be rejected in order to insure against bearing failure. In the jet-propulsion engines, and, earlier, in turbo-superchargers, this was done successfully by injecting solid jets of oil directly into the bearings, the oil striking either on the side of the ball separator or directly onto the balls. Subsequently the oil is thrown off and scavenged out of the bearing housing as rapidly as possible, thereby carrying off the heat and transmitting it to the oil reservoir where it is, in turn, removed by conduction and radiation. On a bearing having a bore of about 110 millimeters (4.3307 inches), as much as three to ten pounds of oil per minute is injected into the bearing. This enables heavy loads to be carried at high speeds.

In a recent test by one of the builders of aircraft engines, a 7224 size bearing was run under about 12,000 pounds thrust at 8000 RPM. Careful observation of temperatures indicated that three pounds of oil per minute was necessary to cool the bearing. Varying the oil flow up to fifteen pounds per minute did not materially decrease the temperature at flows above five pounds per minute. Where jet lubrication of this nature is used, the primary purpose of the solid jet injection is to carry off heat, since very little of this large volume of oil is needed to lubricate the bearing. It has been found to be more effectual than using large volumes of air at low pressure to carry oil mists through the bearing, depending on the volume of air passing through the bearing to cool it.

Lubricating with Grease.

Grease packing of ball bearings is successfully practiced where vertical spindles are involved, as on high-speed jig grinders and electric woodworking shapers, some of the former running as high as 60,000 RPM. Two high-precision ball bearings carry each spindle. In one very successful design applied to internal grinding service, the bearings are grease lubricated from an annular space located above each bearing. At the time of assembly, grease is packed in by hand; and on some machines, there is no provision for relubrication during the running life of the bearing.

Other designs provide for pressure grease-gun relubrication. Lubrication from the original charge of grease is maintained by "bleeding" the grease just sufficiently to maintain a continuous wetting film of lubricant on the bearing parts. Those who favor grease lubrication claim the advantages of low-cost housings, infrequent lubrication, and cleanliness—all important factors in machine design.

Type of Grease to Use.

Certain greases susceptible to shear bleed oil readily and are considered suited for the service mentioned. The presumption is based upon the hypothesis that, since such greases may contain a number of occluded oil pockets, the latter (plus shear susceptibility) tend to maintain a semi-fluid oily condition around the moving parts of the bearing. The fact that the grease is packed into an annular space above the bearing indicates the need for a quick-feeding, medium-consistency grease.

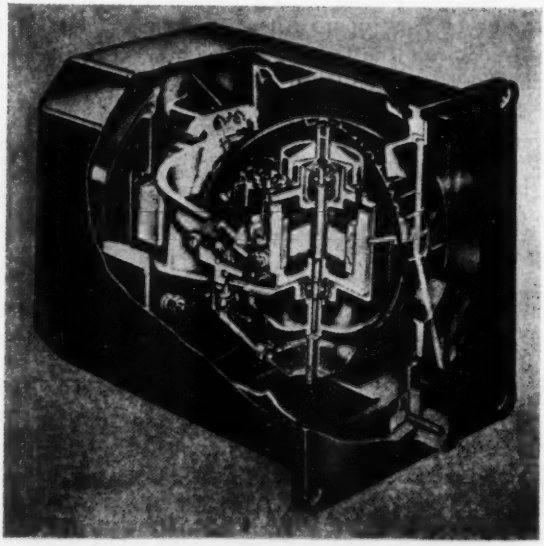


Fig. 7. Section of a Sperry Gyroscope which Operates at Approximately 24,000 R.P.M. The Rotor Bearings are Prelubricated with Grease

Fig. 7 shows a gyroscope which operates at 24,000 RPM. and in which grease is successfully used as a lubricant. Fig. 8 illustrates an assembly of the Moore high-speed grinder head, and shows the annular spaces which serve to retain grease for the upper and lower ball bearings; also the grease seal and labyrinth at the working end of the tool which prevent grinding dust and other non-lubricating material from being drawn up into the bearings.

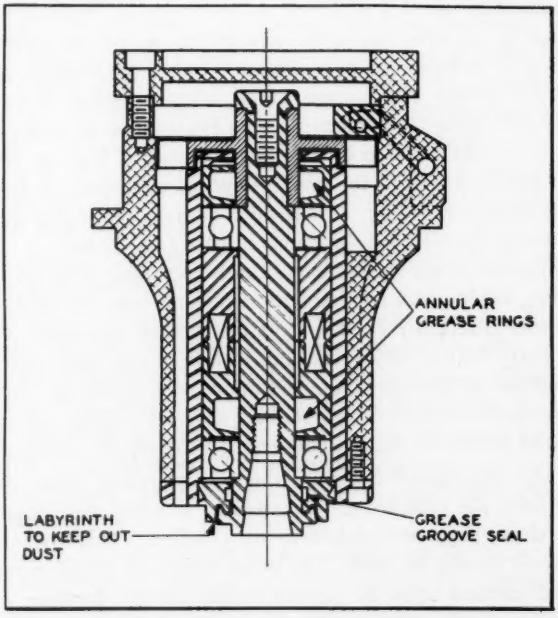


Fig. 8. Cross-sectional Diagram of an Assembly of the Moore High-speed Grinder Head, which Operates at 50,000 R.P.M. Grease is Used for Lubricating the Upper and Lower Ball Bearings

Certain laboratory tests indicate that grease allows a lower temperature rise over ambient than oil. In this regard, the following data is of interest:

Lubricant	Temperature Rise over Ambient, Degrees F.
Spindle Oil with 150 Seconds S.U. Viscosity at 100 Degrees F.	92
Grease with 105 Seconds S.U. Viscosity at 100 Degrees Mineral Oil	25
Grease with 300 Seconds S.U. Viscosity Mineral Oil at 100 Degrees F.	45

Considerable merit is attached to lubrication of high-speed ball bearings by oil mist, since the amount of actual oil subject to churning is reduced to a minimum. Churning results in heat generation within the volume of oil being agitated and consequently increased temperature of the bearing as a whole. This is not a serious problem in bearings running as high as 5000 to 10,000 RPM., provided the unit is exposed sufficiently to the outside air to permit cooling. Where oil is actually circulated through the bearing, the oil itself functions in part as a coolant.

High-speed bearings, however, can be seriously affected by oil churning, the internal friction within causing heat rise and acting as a brake to retard free motion of the bearing elements. This is the primary reason why splash lubrication is not applicable. Oil-mist lubrication prevents these conditions, due to the fact that the relative volume sprayed into contact with the rolling elements is extremely small—in reality, only enough to keep the surfaces wet.

In any high-speed bearing which is provided with oil-mist lubrication, the compressed air used to develop this mist must be as clean, dry, and dust-free as possible. This means that filtered air should be used. The effectiveness of the filtering system will depend upon the source of the air, the location of the compressor intake, the air filters used, the frequency with which they are cleaned, and the possibility of rust and scale forming in air lines due to condensation when the system is shut down. Manufacturers of air-oil mist equipment guard against air contamination by supplying a filter, a moisture remover, and a pressure regulator, so that compressed air from a standard line is properly conditioned before it goes to the spindle.

Minute particles of foreign matter in the air can sufficiently contaminate the oil and penetrate within the bearing to cause

noise and rough rolling. If this continues, obviously, the bearing surfaces may become scored—perhaps so severely as to require replacement. Air-oil mist lubrication has a definite advantage in that the pressure under which it is normally maintained will keep fine grinding dust from being drawn into the bearing. Precision bearings are built with clearances as low as 0.0001 inch. Consequently, any foreign particles above this dimension in the lubricant can cause a rough bearing.

Spindles running in ball bearings at very high speeds cause windage, which is one of the reasons for noisy bearings, according to bearing research authorities. Whether or not this noise is objectionable depends upon the service involved. Obviously, it is desirable to have a quiet bearing by noise testing standards, in the beginning. This is one measure of the suitability of the bearing. In service, it may be different. A grinding spindle or cabin supercharger bearing can produce normal noise without objection. A high-speed bearing in submarine service, however, must run as silently as possible to preclude detection of the craft by enemy sonic instruments.

Dirt is one of the primary causes of bearing noise. Even the cleanest bearing—by manufacturers' standards—will give up some dirt when soaked in a solvent. An ordinarily clean grease contains some foreign particles when studied under the microscope. These can be reduced to a satisfactory minimum by proper handling in the manufacturing process—a usual procedure among those companies that make a practice of supplying high-quality lubricants.

Recently, a sub-committee was appointed by the National Lubricating Grease Institute and the Annular Bearing Engineers committee to study methods for determining the number, size, and nature of foreign particles in greases and to prepare recommendations concerning interpretations of the data in terms of their practical significance.

The low-viscosity mineral petroleum oils so widely used in oil-mist systems can be produced virtually dust-free at the refinery. This, however, does not insure that they will remain so when distributed by the oil-mist equipment unless dust-free air is used.

Methods of Protecting Bearings Against Dirt.

Proper sealing is most important in high-speed bearing operation. Labyrinth seals and slingers have been used to good advantage. When bearings are adequately protected against dirt contamination before, during, and after installation, their life is prolonged and continued, satisfactory operation is more nearly assured. This is particularly true when grease-lubricated bearings are involved. Some spindles may have a tendency to "inhale" after shut-down, others may run with reduced air pressure internally. In dusty atmospheres, this may materially affect bearing operation unless the seals are sufficiently tight and the labyrinths intricate enough to prevent dust being drawn in from the surrounding air.

The problem is not as serious when properly controlled air oil-mist systems are used, where a flow of air through the bearing is continually outward under sufficient pressure to keep fine particles of dust from being drawn in.

The attention given by the anti-friction bearing industry and builders of high-speed machinery to ways and means of insuring positive lubrication is indicative of the importance they attach to this trend toward higher speed operations. It is a fortunate coincidence that either grease or oil can be used satisfactorily, provided the bearings are planned for the specific type of lubrication. Naturally, there are differences of opinion regarding the relative merits of grease and oil, and the essential characteristics.

This article represents a study of the preferred methods of lubrication. Stimulation of interest from a research point of view should result in the development of valuable data concerning the most desirable characteristics of lubricants for high-speed ball bearings.

CONTROLLABLE-PITCH PROPELLERS.

ACKNOWLEDGEMENT.

This article is a brief abstract of the paper presented by Comdr. L. A. Rupp, U.S.N. before the American Society of Naval Architects and Marine Engineers at their annual meeting in Nov. 1948.

Controllable-pitch propellers for ships were first considered seriously in the middle nineteenth century in order to permit feathering the propeller blades when operating with sails only and to provide flexibility when using both sails and steam engines for ship propulsion. A semi-controllable-pitch propeller was even installed in the *Merrimac* (renamed the *Virginia* in her historic encounter with the *Monitor* in 1862.

Most early controllable-pitch mechanisms lacked sufficient ruggedness to withstand the large control forces imposed on them, with the result that failures were frequent. Kaplan turbine-type hydraulic mechanisms developed in Switzerland in 1934 and in Sweden in 1937 proved that controllable-pitch propellers could be made reliable in diverse marine applications. For multi-purpose Diesel tugs, fishing craft, and other types which operate with widely varying towing loads, controllable-pitch propellers permitted utilization of full engine power during all conditions of operation, provided superior maneuvering at slow speeds around docks and when handling trawls and nets, provided greater deceleration in emergency stopping maneuvers, and oftentimes permitted improved fuel economy under cruising conditions. These advantages for towing craft, coupled with the advent of the marine gas turbine, prompted the marine industry in all countries to take renewed interest in controllable-pitch propellers in recent years.

The problem of reversing with a gas turbine installation is not easily solved by conventional means. Astern turbines, such as are commonly used in steam installations, are nearly impossible in gas turbine applications because of excessive windage losses and

dangers from overheating. Reverse gears above 3000 horsepower have not yet been developed and hydraulic reversing couplings, though a possibility, sacrifice efficiency through transmission losses. Available reversing means for larger installations include only electric drive and the controllable-pitch propeller. The former requires greater first cost, weight, space, and transmission losses. The latter has the disadvantages of a possible slight loss in propeller efficiency due to its larger hub size and a more vulnerable location of vital mechanisms, though it has the additional advantage of greater combined turbine and propulsion efficiency at part load by virtue of reducing the propeller pitch and thereby increasing the prime mover speed to a more efficient range.

To evaluate the potentialities of controllable-pitch propeller applications in present day, as well as future prime movers, the U. S. Navy fitted experimental installations in a small Diesel harbor tug in 1940, a 28000-horsepower, twin-screw, turbine-powered World War I destroyer, U.S.S. *Dahlgren*, in 1941, and in a 1000-horsepower Diesel harbor tug, YTB502, in 1946. In addition, over 2400 controllable-pitch propellers, each powered by light weight, non-reversible Diesel engines and absorbing 900 shaft horsepower, were installed in approximately 100 patrol craft and 1100 infantry landing craft which saw action in all theatres of operation during World War II.

Complete trials of YTB502 were conducted in 1946 to determine the centrifugal, frictional, and hydrodynamical blade spindle control forces and to evaluate the free route, towing, dead pull, and emergency maneuvering performance as compared to a sister tug with reversing Diesel engines and a fixed pitch propeller. The controllable-pitch propeller installation had equivalent performance in free route, permitted an increase of over one-third in the towrope pull at dead pull, and stopped in 25 per cent less time and distance in an emergency stopping maneuver. Analyses of various Diesel installations also indicate that fuel savings of 5 to 10 per cent can be effected at cruising power by increasing the pitch and thereby increasing the engine mean effective pressure.

Fig. 1 summarizes the complete comparative performance of the fixed and controllable-pitch installations under all ahead

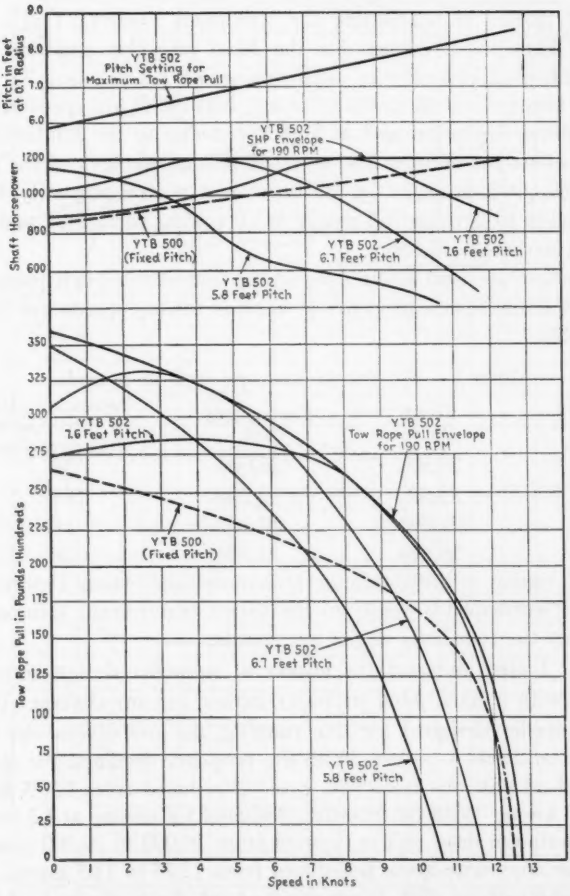


FIG. 1.—COMPARISON OF TOW ROPE PULL AND SHAFT HORSEPOWER OF YTB500 WITH FIXED-PROPELLER AND YTB502 WITH CONTROLLABLE-PITCH PROPELLER AT ALL TOWING CONDITIONS FROM FREE RUNNING TO DEAD-PULL AHEAD

The above results were derived from the standardization, towing, and dead-pull ahead trials of the two vessels, with all values adjusted to a common base so that the overload torque condition corresponding to 1,200 horsepower at 190 revolutions per minute is not exceeded and the revolutions per minute are limited to 190. The upper curve indicates the optimum pitch setting for maximum tow rope pull for YTB502.

operating conditions. The towrope pull envelope for the controllable-pitch propeller represents the maximum available towrope pull for 1200 shaft horsepower and 190 revolutions per minute when the pitch is adjusted in accordance with the upper

curve, labeled "Pitch Setting for Maximum Towrope Pull". The shaft horsepower curves for the fixed propeller and the controllable-pitch propeller at pitch settings of 5.8, 6.7, and 7.6 feet show clearly how the available power falls off at low speed due to the torque limitation and at high speed due to the limitation in revolutions per minute. The towrope pull advantages of the controllable-pitch propeller in the lower plot accrue primarily from the ability to develop full power at all towing conditions and not from increased efficiency.

The towrope pull advantages of the controllable-pitch propeller over the fixed-pitch propeller at various towing speeds are listed in Table 1.

Table 1. Maximum towrope pull in pounds.

Towing speed in knots	Fixed Pitch	Controllable Pitch	Percentage gain for controllable-pitch propellers
0	26,500	36,000	36
4	23,800	32,200	35
7	21,000	28,100	34
10	16,700	21,400	28

The variation in the maximum free-route speed is not significant, since it corresponds nearly to the thrust requirement differences between the two hulls at the top speeds.

Fig. 1 also indicates the effects of propeller design compromises with a fixed-pitch propeller harbor tug installation. With the propeller designed for free running, the loss of towrope pull shown in Table 1 occurs. With the propeller designed for maximum dead pull, the free-route speed is reduced from 12.75 knots to 10.7 knots. With the propeller designed for towing at 4.5 knots, the maximum dead pull is reduced from 36,000 to 30,800 pounds and the free-route speed is reduced from 12.75 to 11.7 knots.

Charts such as Fig. 1 are very useful for towing operations with controllable-pitch propeller tugs, since the tug master can determine at a glance the maximum towing speed for a given towing load and the optimum pitch setting to attain that speed.

Starting with approximately the same initial speed and power, the transient values of speed, reach, torque, thrust, and revolutions per minute for a complete emergency stopping maneuver of YTB500 and YTB502 are plotted in Fig. 2. The fixed-pitch pro-

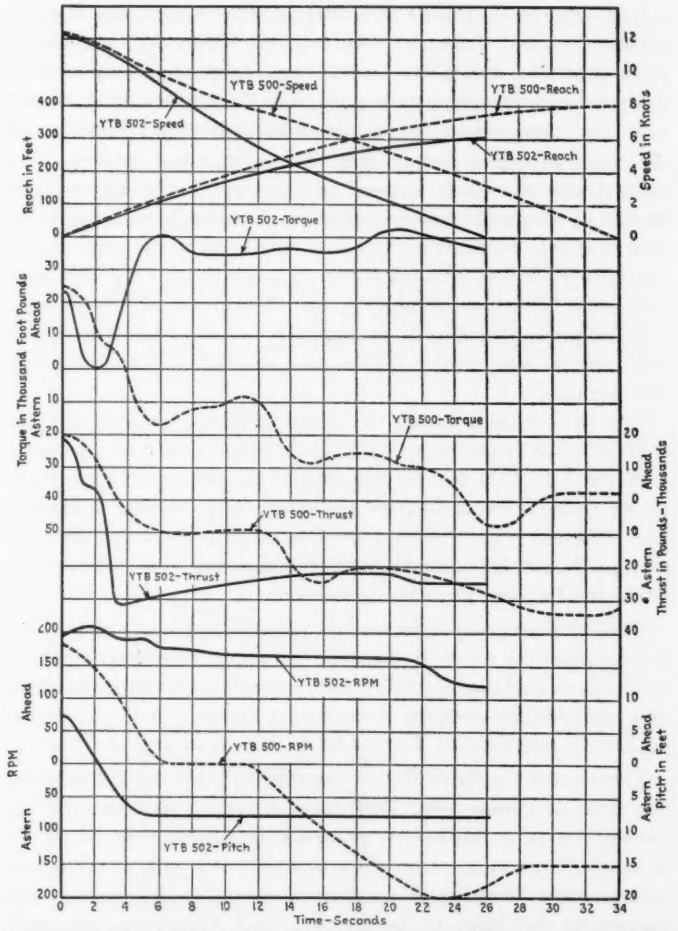


FIG. 2 —COMPARISON OF SPEED, REACH, TORQUE, THRUST, AND REVOLUTIONS PER MINUTE DURING AN EMERGENCY ASTERN MANEUVER FROM DESIGNED SPEED AHEAD FOR YTB500 WITH FIXED-PITCH PROPELLER AND YTB502 WITH CONTROLLABLE-PITCH PROPELLER

peller reversal with YTB500 required 34 seconds and a reach of 402 feet to bring the tug to zero speed, while the controllable-pitch propeller in YTB502 accomplished the maneuver in 26 seconds with a reach of 303 feet.

The differences in time and distance traversed before stopping can be explained from the thrust curves. With the controllable-pitch propeller, full pitch reversal was accomplished in about 5 seconds and maximum astern thrust was developed only 4 seconds after the command to execute the maneuver. In YTB500, with the fixed-propeller installation, the mechanical brakes were applied and the shaft stopped in 6 seconds; the engines were started in reverse 11 seconds after the command, with the result that astern thrust equivalent to that developed by the controllable-pitch propeller was attained about 15 seconds after the execution command. The considerably greater astern thrust of the controllable-pitch propeller in the first 15 seconds of the maneuver accounts for the greater deceleration and smaller stopping time and distance.

The shaft torque during the stopping maneuver with the controllable-pitch propeller in YTB502 dropped to zero in 2 seconds, while in YTB-500 the reverse torque did not reach appreciable values until 16 seconds after the emergency reversal command. The temporary reverse overspeeding of the fixed propeller between 20 and 26 seconds after commencing the maneuver is attributed to propeller blade cavitation.

Further emergency astern trials of YTB502 were carried out to determine the effect of changing the pitch more slowly, but the results indicated that for this class of vessel the greatest deceleration and shortest stopping distance are obtained with the fastest pitch reversing time of 5 seconds, as shown in Fig. 2.

These results indicate that a controllable-pitch installation with any type of prime mover will have transient maneuvering advantages over a fixed-pitch propeller installation, primarily because the large momentum changes associated with stopping and reversing the direction of rotation of propeller and shaft system masses do not have to be effected, and, therefore, maximum astern thrust can be attained more rapidly.

The *Dahlgren* trials in 1941 indicated that considerable losses can result from large propeller hubs with improper fairing forward and aft of the propeller, but other tests and installations indicate that losses from properly faired controllable-pitch propeller hubs need never exceed 1 to 2 per cent. The *Dahlgren* trials

showed that turbine-driven vessels with controllable-pitch propellers can be stopped in less than two-thirds of the distance as compared to reversing turbines and fixed propellers, and that cruising fuel savings of 10 to 20 per cent can result from using decreased pitch and corresponding higher turbine speeds.

The control rod forces required to overcome the friction of the mechanism were determined by measuring the transient servomotor pressures during an emergency pitch-changing cycle with

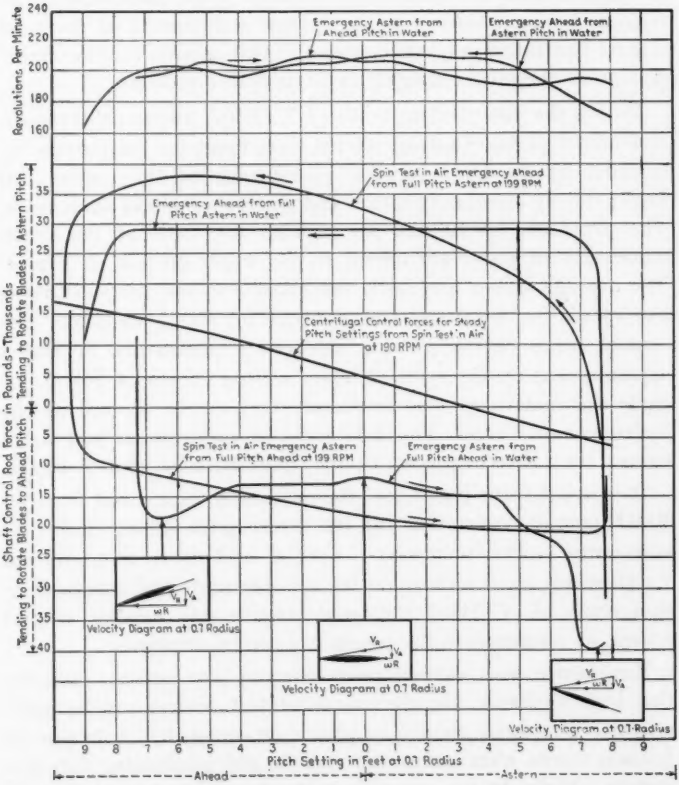

Revolutions Per Minute
240
220
200
180
160
Emergency Astern from Ahead Pitch in Water
Emergency Ahead from Astern Pitch in Water
Shaft Control Rod Force in Pounds-Thousands
Tending to Rotate Blades to Astern Pitch
Tending to Rotate Blades to Ahead Pitch
35
30
25
20
15
10
5
0
5
10
15
20
25
30
35
40
Spin Test in Air Emergency Ahead from Full Pitch Astern at 199 RPM
Emergency Ahead from Full Pitch Astern in Water
Centrifugal Control Forces for Steady Pitch Settings from Spin Test in Air at 190 RPM
Spin Test in Air Emergency Astern from Full Pitch Ahead at 199 RPM
Emergency Astern from Full Pitch Ahead in Water
Velocity Diagram at 0.7 Radius
Velocity Diagram at 0.7 Radius
Velocity Diagram at 0.7 Radius
9 8 7 6 5 4 3 2 1 0 1 2 3 4 5 6 7 8 9
Pitch Setting in Feet at 0.7 Radius
Ahead
Astern

FIG. 3.—COMPARISON OF CONTROL ROD FORCES MEASURED DURING EMERGENCY ASTERN AND EMERGENCY AHEAD PITCH-CHANGING CYCLES FOR THE CONTROLLABLE-PITCH PROPELLER OF YTB502 OPERATING IN AIR AND IN WATER

The velocity diagrams in the lower views illustrate the flow conditions for the propeller blades at the 0.7 radius and at the indicated pitch settings during an emergency astern maneuver from designed speed ahead.

the propeller spinning at constant revolutions per minute in air. The derived control rod force for emergency astern and emergency ahead maneuvers in air at 199 revolutions per minute are plotted in Fig. 3, with arrows indicating the direction of pitch-changing motion. The centrifugal components of the control rod force for steady pitch settings at 190 revolutions per minute are plotted in Fig. 3. The difference between the measured forces during the emergency ahead and astern pitch-changing cycles and the centrifugal control forces for steady pitch settings represents the approximate frictional control forces of the mechanism. The friction forces proved to be the largest components of the total control rod forces, averaging about 25,000 pounds for the major portion of the pitch-changing cycle in both directions.

During the maneuvering trials of YTB502, additional transient servomotor pressure measurements were made for the purpose of determining the hydrodynamic components of the control rod forces during emergency astern and emergency ahead maneuvers. The propeller revolutions per minute for both of the crash maneuvers in water are shown in the upper portion of Fig. 3. The average values are fairly comparable to the propeller rotative speeds for the steady and maneuvering spin tests in air. The control force curves for the emergency maneuvers in water, superimposed on the spin test maneuvering curves of Fig. 3, include the centrifugal and frictional components, as well as the hydrodynamic forces. The differences between the air and water curves are a direct measure of the hydrodynamic control forces.

It is noted from Fig. 3 that the maximum control rod force of 40,000 pounds occurred at the full astern pitch setting during an emergency astern maneuver. Control rod force measurement for all steady pitch settings during the free-route and standardization trials of YTB502 indicated considerably smaller control forces, as compared to the emergency astern maneuver.

Experience with various pitch-changing mechanisms indicates that both hydraulic and electro-mechanical devices can be made reliable. Emphasis should be placed on simplicity, reduction in friction forces, elimination of backlash, and minimizing hub diameters. Methods are now available for achieving the proper blade designs for controllable-pitch propellers in order to attain

maximum overall efficiency and minimum control force requirements for the mechanism.

The appreciable advantages of controllable-pitch propellers in providing increased towing power, flexibility, maneuverability, and fuel economy for tugs, fishing craft, and other vessels which operate at variable loads, as well as for providing flexibility and a convenient reversing means for marine gas turbine power plants, point the way to continued developments and increased applications for future ship propulsion.

A LARGE MARINE OIL-OPERATED REVERSE-REDUCTION GEAR.

ACKNOWLEDGMENT.

This article is reprinted from "The Shipbuilder and Marine Engine Builder" for August 1948.

The running test of what is claimed to be the largest marine reverse-reduction gear so far manufactured was recently witnessed by our editorial representative at the Slough works of Modern Wheel Drive, Ltd., by whom the set was designed and manufactured. This installation, a development of the SLM reverse-reduction gear, has been constructed for the motorship *Catford,* a collier built for the service of the South Metropolitan Gas Company, by Messrs. S. P. Austin & Son, Ltd., Sunderland.

The gear, illustrated in Figs. 1 and 2, is of the double-reduction type, and is capable of transmitting 2000 HP., with an input-shaft speed of 450 RPM., although, in the *Catford,* it is required to transmit only 1425 HP. at an engine speed of 423 RPM.

As will be seen from the illustrations, the primary, or input, shaft—connected to the engine by means of a Bibby coupling—carries the ahead and astern first-reduction pinions. The first ahead pinion meshes directly with the first-reduction gear wheels, but the astern pinion drives the first astern reduction gear wheels through an idler.

Incorporated in the first-reduction wheels are the oil-operated clutches, which are engaged by oil pressure through a hand-operated control cock. When the control cock is in the neutral position, the clutches are disengaged, and the first-reduction wheels rotate freely on their shafts. The gears are always in mesh, and when the control is in, say, the ahead position, the astern first-reduction wheels rotate freely, and *vice versa.* Integral with the output shaft is the thrust collar, the thrust of the propeller being taken by a Michell-type thrust block, the housing of which can be seen in the foreground of Fig. 2.

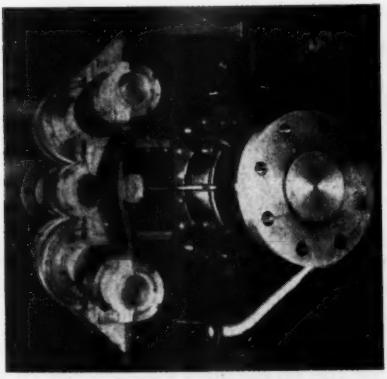


FIGURE 2.—GEARBOX WITH TOP COVER REMOVED, LOOKING FORWARD.

FIGURE 1.—STARBOARD VIEW OF SLM OIL-OPERATED REVERSE REDUCTION GEAR WITH TOP COVER REMOVED.

At the normal engine speed of 423 RPM., the propeller speed ahead is 80 RPM., and astern 82 RPM., giving an ahead reduction ratio of 5.42 : 1, and an astern ratio of 5.28 : 1. Double-

helical gearing is employed throughout, and the entire unit is of sound design and robust construction.

For the purpose of the test at the Slough works, the gear was coupled to a 550-HP. variable-speed A.C. motor running at 1200 RPM. and arranged to drive the input shaft at 423 RPM., giving a speed of 80 RPM. at the output shaft, which, in turn, was coupled, through speed-increasing gear, to a Heenan & Froude dynamometer. The actual brake load on the dynamometer was only 100 HP., as a larger dynamometer was not available. Preceding the shop test of the assembled unit, individual clutch tests were carried out, each clutch having been subjected to a full-load power test of two hours' duration. These individual tests showed that the oil-operated clutches had an ample margin of safety, their maximum torque-carrying capacity being, in fact, 45 per cent above the full-load torque which they are called upon to transmit under actual service conditions.

A smaller gear box of this type was recently returned to the works for examination, survey and overhaul, after about eight years' service; it is of interest to record that it was unnecessary to carry out any repair or adjustment—a circumstance which indicates very convincingly the sound design and robust construction of the gears and the clutch.

The shop test, within the capacity of the available dynamometer, was entirely satisfactory; in particular, the changes from ahead to astern, and *vice versa,* were performed in a matter of seconds. When the control lever was moved from the neutral to the ahead or astern position, the gear picked up almost immediately; and, on the lever being restored to the neutral position, the gear was rapidly brought to rest, the response being smooth and quiet, and a pleasure to observe.

To demonstrate the flexibility of the gearing, the change in the direction of rotation was carried out without any pause at the neutral position, although, naturally, this would not be considered desirable on shipboard, with "way" on the vessel. The gear control is interconnected with the engine governor, and, as soon as the control lever is moved to change the direction of the rotation, the engine speed is automatically reduced; thus, the actual re-

versal of the propeller and shafting is effected at the idling speed of the engine.

The *Catford,* for which this interesting gear unit has been designed, is 304 ft. in length and 44 ft. 3 in. in breadth, and she has a carrying capacity of 4100 tons on a draught of 19 ft. 2 in. The prime mover is a Ruston four-stroke cycle, six cylinder Diesel engine, and the power is sufficient for a speed of 10.5 knots. The vessel will be employed in the Thames-North-East Coast service, and her performance will be watched with much interest by marine engineers generally.

Modern Wheel Drive, Ltd., have also been entrusted with an order for a reverse-reduction gear for a twin-engine installation, each prime mover of which develops 1400 HP. at 300 RPM., the output-shaft speed being 165 RPM. In this gear, provision is made to isolate either engine at will, and also to start one engine from the other by engaging the appropriate oil-operated clutches.

Further designs for a very wide field of application are also in hand, including the undermentioned units:—

(*a*) A gear unit, for ahead and astern running, for a steam-turbine drive of 24,000 HP. at 6000 RPM., with a reduction ratio of 17.2 to 1;

(*b*) A similar unit for a gas-turbine drive, the input-shaft speed being 12,000 RPM. and the reduction ratio 34.3 to 1; and

(*c*) A gear unit (incorporating ahead and astern elements), coupled to a turbine developing 15,000 HP. at 3000 RPM., and having a reduction ratio of 12:1.

DISCUSSION BY MR. PETER W. MOURER, JR., OF THE PAPER "METALLURGICAL TRACERS AND NAVAL RESEARCH" BY JULIUS J. HARWOOD. FEBRUARY 1948 (VOL. 60 NO. 1.)

This paper, by a very excellent and scholarly technique, brings forth a field which should find immediate and practical application by all engaged in the production of ferrous and other alloys. The writer, resident in a region only beginning its industrial development, but possessed of not inconsiderable resources of high-grade iron, magnesium, chromite, copper, molybdenum, manganese, aluminum and other strategically important minerals, coupled with a rapidly increasing development of non-deteriorating potential of hydro-electric power, is particularly interested in the application of such procedures as applicable to the industrial development of the Pacific Northwest region both as a local good and as being in harmony with the strategic necessity of industrial dispersion.

Development of the Puget Sound naval installations in an orderly and increasingly efficient manner is closely interrelated to maintenance of privately-owned heavy industrial facilities of this region on a gradient of constantly increasing expansion. During the last "hot" war, in which remote manufacture of components and centralization only at assembly was brought, through necessity, to a high state of perfection, we were constantly plagued with logistical problems arising from the concentration of industrial development in the eastern portion of the country. This condition naturally prevailed increasingly as our effort became concentrated in the Pacific.

In view of the present international situation, it is notable that research, apparently confined mainly to ferrous alloys at present, as indicated by the subject article, has been in eastern universities and laboratories. Would it not seem fitting, in view of the expanding industrial facilities of the West and the location of a major isotopic production plant here, that future research, particularly in the vital field of light metals and their alloys, might be contracted to universities and technical schools in the Pacific North-

west? Release of the knowledge so gained, under appropriate security restriction, to industrial plants of this area, might do much to solve in advance the logistical difficulties above referred to.

REPLY TO DISCUSSION OF MR. PETER W. MOURER, JR.

By The Author—Mr. Julius J. Harwood.

Mr. Mourer rightfully points out, from strategic and logistic considerations, the desirability of not concentrating our research effort in any one sector of the country. I wish to take this opportunity to assure Mr. Mourer that the overall research program of the Office of Naval Research extends into academic, private and industrial research laboratories of all geographic areas and practically every State of the union.

It is merely happenstance that the two programs described in the article "Metallurgical Tracers and Naval Research" are being conducted at Eastern Universities. These two projects were among the earliest to utilize radioactive isotopes for metallurgical research and it is hoped that the application of "tracers" in metallurgical research will become more widespread as the techniques and handling procedures become more widely known. Studies employing radioisotopes in several fields of research are currently underway in numerous other research laboratories, including some in the far west.

It may be of interest to know that the ONR fundamental research program is the result of voluntary, unsolicited proposals from scientists seeking support for conducting research projects in which they are interested. The metallurgical portion of that program is concerned with all phases of physical metallurgy and it is hoped that the information which will be forthcoming will be applicable to all metallurgical problems and not merely confined to particular alloy systems. Space does not permit a description of our entire metallurgical program at this time, but suffice it to say that problems involved with non-ferrous and light metals are not being neglected. Publication of the results of ONR research projects in the recognized scientific and technical journals is encouraged so that the information will be readily available.

DEATH NOTICES.

The Society has learned with regret of the deaths of the following members. Time and facilities have prevented the preparation of individual obituary notices.

Naval Members.

Captain Lybrand P. Smith, U.S.N., Ret.
Commander Miles Joyce, U.S.N., Ret.
Ensign Rufus Luke André, U.S.N.

Civil Members.

James L. Betts.
Robert J. Broege.
William H. Collins.
James A. Dempsey.
J. E. Peterson.

Associate Members.

Carl Henry Graesser.
Sigmund Hejke.
C. A. Longstreth.

NEW MEMBERS.

It has been the privilege of the Society to accept the following new members since the publication of the list in the November, 1948, JOURNAL.

NAVAL.

Brookman, Stanley M., Ensign, U.S.N.R.,
Marine Engineer, Union Oil Co. of California,
Mail: 652½ W. Huntington Drive, Arcadia, Calif.

Bushey, Arthur Clifton, Jr., Commander, U.S.N.,
Code 440, Bureau of Ships, Navy Dept., Washington, D. C.

Caine, Martin G., Lieutenant, U.S.N.R.,
Plastics Division, Monsonto Chemical Co.,
Springfield, Mass.

Callahan, John W., Lieutenant, U.S.N.R.,
Student, A & M College of Texas,
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Dermody, Eugene Thomas, Chf. Mach., U.S.N.R.,
Engineer, Chicago Milwaukee & St. Paul R.R.,
Mail: 1240 A North 45th St., Milwaukee 8, Wis.

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Washington, D. C.

Forhan, Edwin T., Commander, U.S.N.R.,
Special Tester, Cleveland Diesel Engine Division,
Mail: 1306 Belle Ave., Lakewood 7, Ohio.

Forest, Francis X., Captain, U.S.N.,
David Taylor Model Basin, Washington 7, D. C.

Fossum, Theodor T., Ensign, U.S.N.
U. S. Naval School, U. S. Naval Base, Newport, R. I.

Konen, Gerald James, Ensign, U.S.N.R.,
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901 So. 20th Ave., Yakima, Wash.

Mandeville, A. C., Lieut. Commander, U.S.C.G. (Ret.),
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Mail: 3402 Giles Place, New York 13, N. Y.

Raven, Richard K., Lieutenant, U.S.N.R.,
Supervisor of Drafting, Bath Iron Works, Inc.,
Mail: 449 Middle St., Bath, Maine.

Reynolds, James R. Z., Commander, U.S.N.,
Box 27, Submarine Base, New London, Conn.

Searle, Willard F., Jr., Lieutenant, j.g., U.S.N.,
U.S.S. *Weiss* (APD 135), c/o Fleet P.O., New York, N. Y.

Seebar, Arthur F., Lieut., j.g., U.S.N.R.,
Service Engineer, Steam Turbine Div.,
Westinghouse Electric Corp.,
Mail: 200 North Jackson St., Media, Pa.

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230 75th St., Brooklyn 9, N. Y.

Wallen, Norman Edward, Ensign, U.S.N.,
U.S.S *Cone* (DD 866), c/o Fleet P.O., New York, N. Y.

Wertz, Francis D., Lieut., j.g., U.S.N.R.,
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Whitaker, Francis H., Captain, U.S.N.,
Staff, Pacific Reserve Fleet, Treasure Island,
San Francisco, Calif.

Wilson, Harrel M., Lieut. Commander, U.S.N.R.,
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Carlson, Bergiton, Supervisor, Hull Engineering,
Gibbs & Cox, New York,
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Cary, George F., Hull Engineer, Bath Iron Works, Inc.,
Mail: P.O. Box 301, Bath, Me.

Egli, Adolf, Professor of Mechanical Engineering,
University of Pennsylvania, Philadelphia 4, Pa.

Elson, William C., Manager, Ordnance Sales, Aircraft,
Federal and Marine Divisions, General Electric
Company, Schenectady, N. Y.

Kane, Joseph J., Estimating Engineer, Federal Shipbuilding
& Dry Dock Co.,
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Kumpf, August W., Engineer, American Engineering Company,
Aramingo Ave. and Cumberland St., Philadelphia, Pa.

Low, Arthur Frederick, N.A.,
230 B Dewey Way, Honolulu, T.H.

Martin, Roger H., Manager, Marine Section, Westinghouse
Electric Corporation, Sturtevant Division,
Hyde Park 36, Mass.

Mongon, William J., Engineer, I.M.O. Pump Division
De Laval Steam Turbine Co.,
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Morris, E. T., Jr., Division Manager, Electronics & X-Ray Division
Westinghouse Electric Corporation,
2519 Wilkins Ave., Baltimore 3, Md.

Morse, Samuel S., Asst. Naval Architect & Marine Engineer
c/o Atlantic Refining Company,
260 South Broad St., Philadelphia, Pa.

Naab, Julius Nash, Engineer in Charge Compressors,
Ingersoll Rand Co.,
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Noltein, George F., Chief Engineer, National Supply
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Sabo, Benjamin J., Liason Rep. for Gibbs & Cox, Inc. at
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Thummim, Carl Engineer in Chief, Development,
I.T.E. Circuit Breaker Co.
Mail: 909 MacDade Boulevard, Lansdowne, Pa.

Tukey, Egbert O., Chief Engineering Draftsman, Bath Iron
Works, Inc.,
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Yourch, Anthony, Chief Marine Engineer, George P. Wagner
Associates,
New York,
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ASSOCIATE.

Brown, Buford M., Asst. Manager, Transportation Sales Dept.
Westinghouse Electric Corporation, East Pittsburgh, Pa.

Hall, Gerald B., Diesel Field Engineer, Baldwin Locomotive
Works,
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Kuhns, William O., Student Engineer, Cleveland Diesel Engine Division,
Mail: 1426 Belle Ave., Lakewood, Ohio.

McClees, Harry L., President, Crucible Steel Casting Company, Landsdowne, Pa.

Paquette, Armand L., Manager Aviation Sales, Westinghouse Electric Corporation, East Pittsburgh, Pa.

Patterson, William C., Lieut. Commander R. Canadian Navy, H.M.C.S. *Ontario*, c/o F.M.O. ESQUIMALT, B.C., Canada

Palmer, M. S. Manager, Hancock Valley Division, Manning, Maxwell & Moore, Inc.,
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Simmons, L. G., Sales Manager, ADE Co., Chicago, Ill.,
Mail: 4416 N. Hermitage Ave., Chicago 40, Ill.

Vor, Ir. Teunis Ary, Lieut. Commander (E) Royal Netherlands Navy, Engineer in Chief Dept. of New Construction, Navy Dept., The Hague,
Mail: Binckhorstlaan 26, Voorsburg, Netherlands.

ASSOCIATION NOTES.

Original Articles.

The editor of the Journal is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
Bureau of Ships, Navy Department
Washington 25, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Addi-

tional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Subject Matter and Authors.

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries

(*a*) What subject would you, as a member, like to see covered by an article in the Journal.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

Photographs.

Photographs of current or historical interest to readers of the Journal are desired. Any which are accepted by the Society will be puchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

Notice of Deaths.

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

Election of Officers.

The following officers were elected by vote of the Society for the years 1949-1950:

President:
Rear Admiral T. A. Solberg, U. S. Navy

Secretary-Treasurer:
Captain J. E. Hamilton, U. S. Navy

Member of Council (term ending Dec. 31, 1950):
Captain Homer Ambrose, U. S. Navy
Captain T. T. Dantzler, U. S. Navy
Captain G. F. Hicks, U. S. Coast Guard
Commander Nicholas Tiedeman, U. S. Naval Reserve
Mr. R. M. Wilson

The new officers took office on 1 January, 1949.

Annual Banquet.

Plans have been completed for the Annual Banquet, to be held at the Hotel Statler, in Washington, D. C., on Friday, 29 April, 1949. Formal reservation blanks have been issued.

The November 1948 Journal was in error in stating that Mr. C. E. Wilson, who will make the principal industry address, was chairman of the Board of General Electric Company. He is President of General Electric.

Vice Admiral Robert B. Carney, U.S.N. will be the principal Naval speaker.

Addresses.

Once more attention is invited to the necessity of keeping the Society advised as to changes in addresses.

Meeting.

The Spring Meeting of the Society for Experimental Stress Analysis will be held at Hotel Statler, Detroit, Michigan on May 19, 20, 21, 1949. Inquiries should be addressed to the Society for Experimental Stress Analysis, P. O. Box 168, Cambridge 39, Mass.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.

ERRATA.

"Propeller Characteristics" published in the November 1948 issue (Vol. 60, No. 4).

Page 463, second line from bottom, inches symbol should be after 2 (11.62″) instead of (11″.62).

Page 466, line 14, change $J = \frac{vn}{d}$ to $J = \frac{v}{nd}$.

Page 472, line 5, substitute 'above' for 'below'.

Are you up on your tows?

Today, more and more towboats are pushing bigger loads at lower cost with General Motors 2-cycle Diesel engines. These dependable engines deliver maximum power at minimum cost—provide high performance with low maintenance. Backed by 37 years' experience, GM Diesel engines are your best answer for towboat power.

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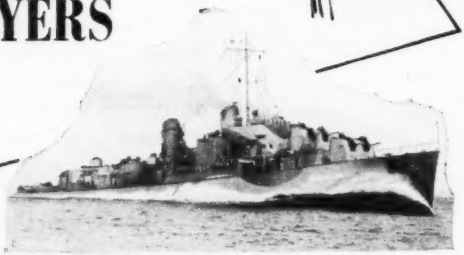

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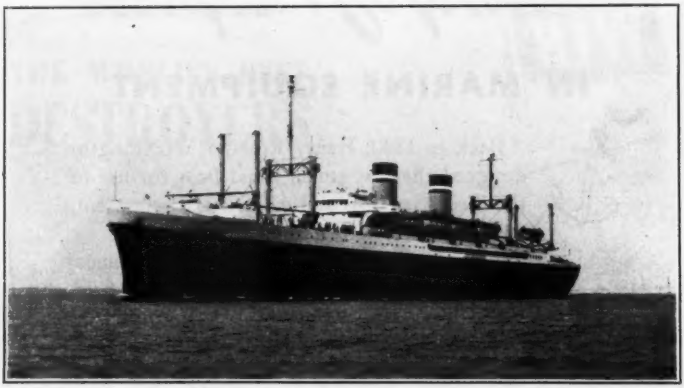

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The GENERAL DANIEL I. SULTAN, third of the P-2 Army Transports converted to furnish modern and comfortable overseas transportation to members of our Armed Forces. Two other ships of this Class are undergoing conversion at Newport News.

These vessels were five of the twenty-one major ship conversion contracts received by Newport News since the war's end.

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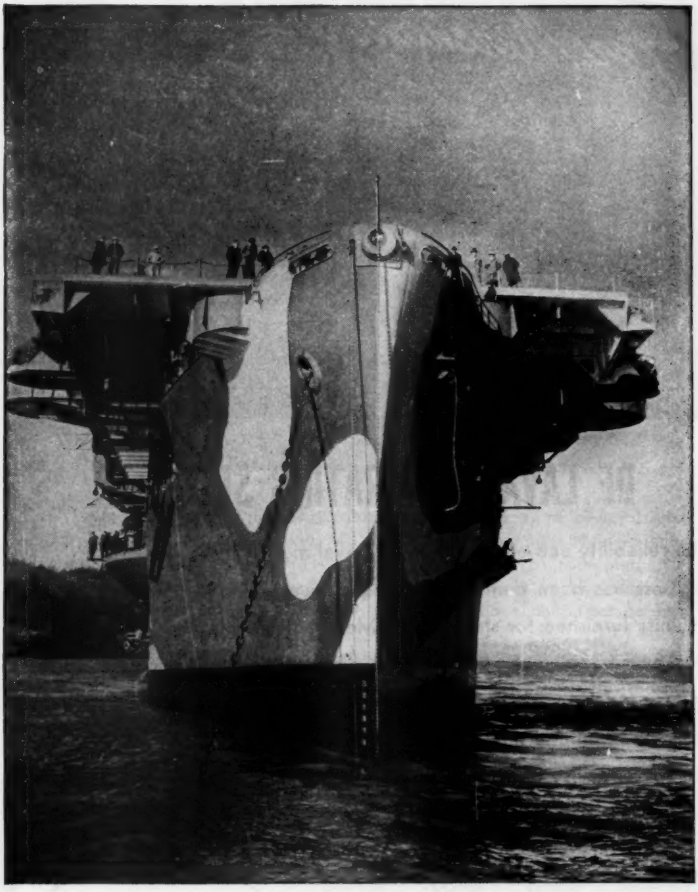

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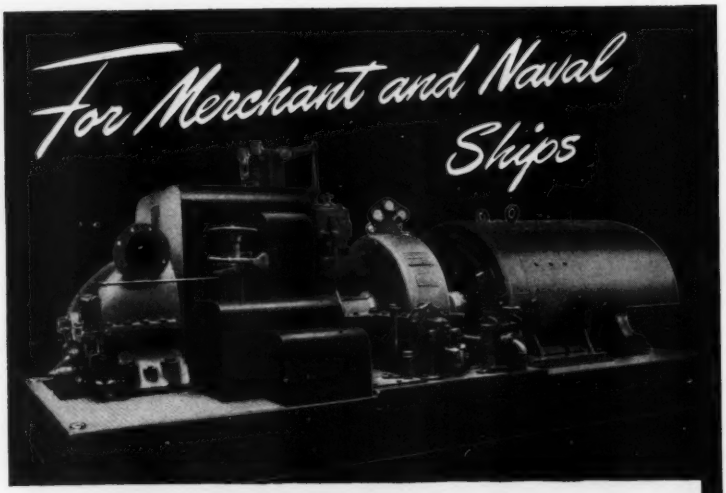

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The reliability and efficiency of De Laval marine turbo-generating sets has been demonstrated conclusively by the hundreds of units furnished for ships of the Merchant Marine and Navy. De Laval turbo-generating sets are distinguished by their neat compact design, accessibility of all parts, simplified governor control, automatic nozzle control and other features contributing to reliability, efficiency and smooth operation.

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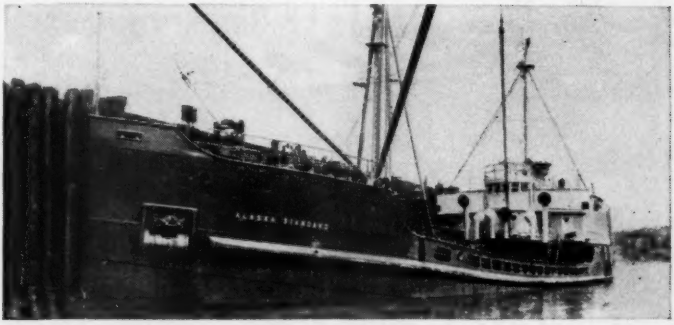

A fine example of the trouble-free durability of G-E marine propulsion drives is the case of the 210-foot tanker MS *Alaska Standard*. For 24 years this "floating bulk plant" has supplied petroleum products to fishermen, fur trappers, and storekeepers along the southeastern Alaska coast. Although it has logged close to a million miles, not once have major repairs been needed on her G-E diesel-electric drive!

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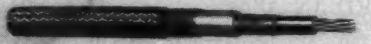

(a) Single conductor rubber insulated cable

(b) Multi-conductor Navy Type power cable

(c) Varnished cambric marine cable with basketweave armor

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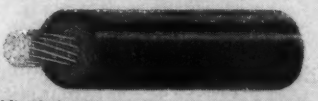

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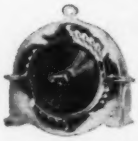

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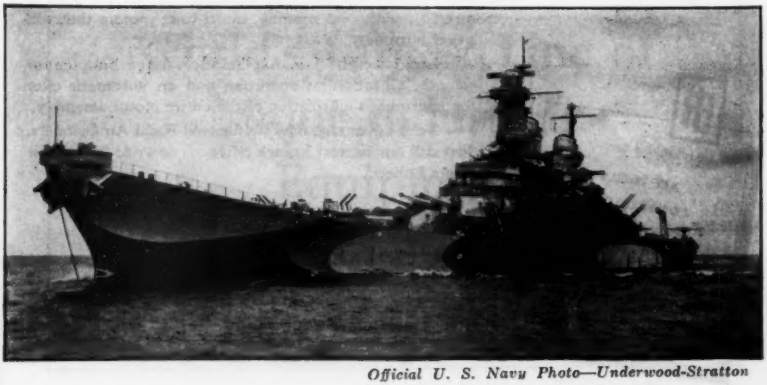

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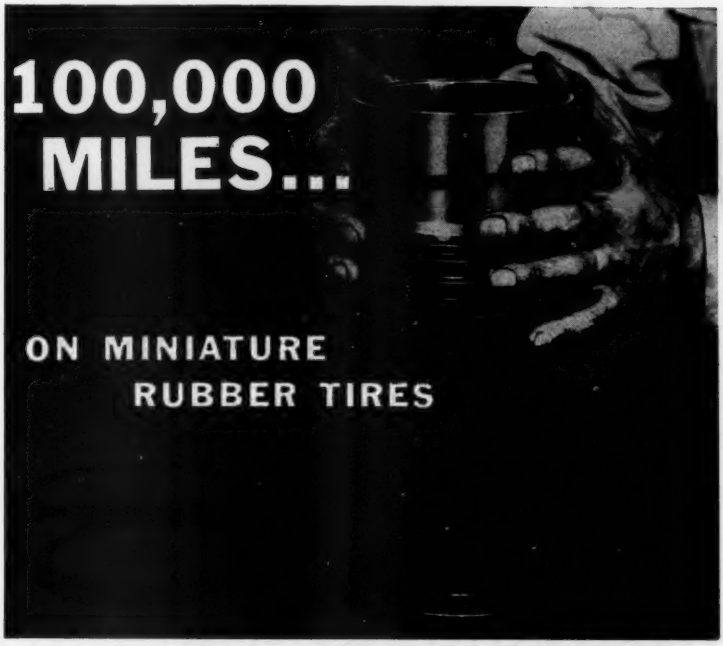

Photograph courtesy of Cummins Engine Company, Inc., Columbus, Indiana

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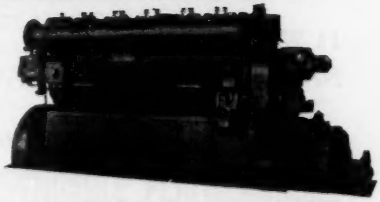

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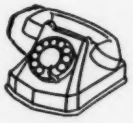

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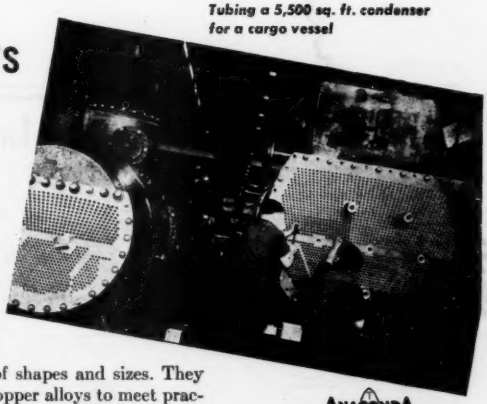

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